MACHINERY.

January, 1908.

DESIGN OF ROTATING DRUMS.

ULRICH PETERS.*

THERE are many materials, such as marls, sand, clay, ore, phosphate rock, gypsum, cement, etc., used for different manufacturing purposes, which require to be either dried, roasted, or calcined, or freed from the water chemically bound. Usually such materials are freed from their moisture at the mine or factory, in order to save the unnecessary expenses of shipping water. Technically, this drying, evaporating, burning up, or roasting is most quickly performed in the so-called rotary dryers or kilns. This dryer, or kiln, consists of a slowly-rotating cylindrical tube or drum, which not seldom has a length of over 100 feet and a diameter up to 8 feet. The axis of rotation of these dryers is not always horizontal, often being more or less inclined. The shell usually rests on rollers, and the inside shell, according to requirements, either is provided with cascade bars (see Fig. 1), which lift and distribute the material over the entire space, or may be brick lined, or left bare.

By moving the supports toward the ends, making $C=0$, Fig. 3, the equation reduces itself to

$$M_x = \frac{W+U}{2}\left(\frac{x^2}{L} - x\right) \qquad (2)$$

A maximum bending moment is produced when $x = \frac{L}{2}$, therefore

$$M_b = -\frac{W+U}{8} L \qquad (2a)$$

Similarly by concentrating both supports A and B in the middle, $C = \frac{L}{2}$, (Fig. 4):

$$M_x = \frac{W+U}{2}\left(\frac{x^2}{L} - x + \frac{L}{2}\right) \qquad (3)$$

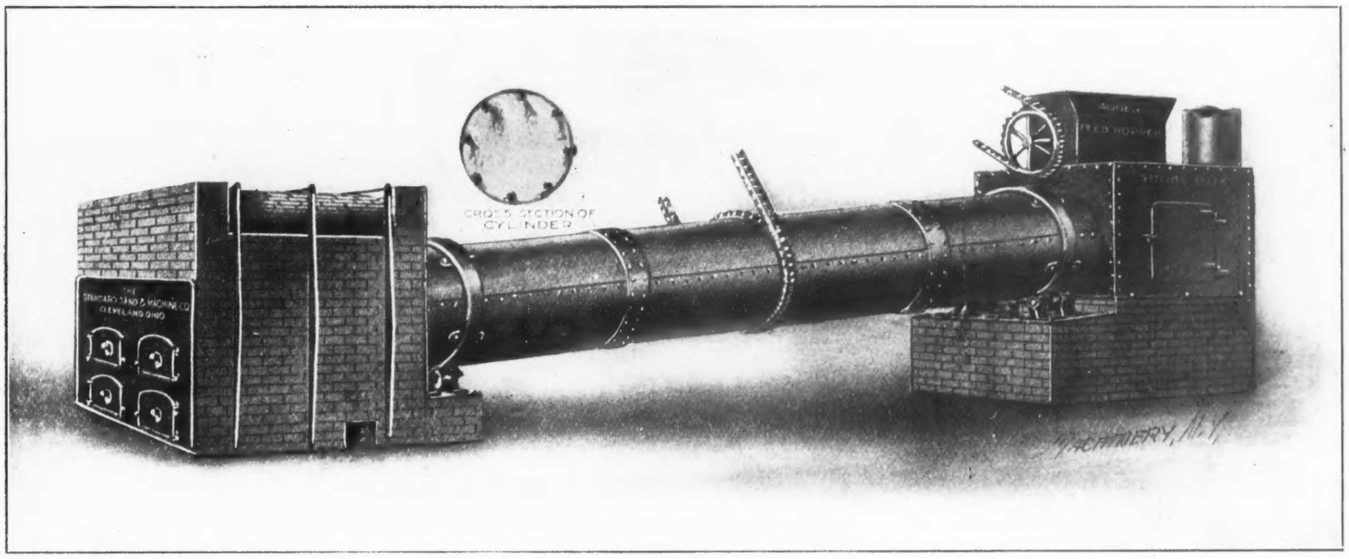

Fig. 1. General Appearance of Drying Drum.

Large foundry tumblers are used for the purpose of smoothing and polishing the surfaces of various products, such as castings, tubes, etc., and similar tumbling barrels are also used in laundries. These drums or barrels, when not charging and discharging through the ends, are provided with suitable openings, located on the side of the drum, for the purpose of charging and removing the material. Finally, there are a great many revolving mining screens which have a close technical relation to the above drums, so far as construction is concerned. The object of this article is to discuss the principal points considered in calculating the strength and proportion of rotating cylinders.

In the following:

W = total weight of drum in pounds,
U = total weight of the uniformly distributed material,
L = length of the cylindrical drum, in feet,
C = cantilever arms, in feet,
M = moment; M_b = bending moment; M_t = twisting moment.
R = moment of resistance, usually called section modulus,
I = moment of inertia,
D = outside diameter, d = inside diameter of drum,
S = unit stress in material.

In general, a beam supported at two points, A and B, located at equal distances C from the end, Fig. 2, will have anywhere at the distance x the bending moment.

$$M_x = \frac{W+U}{2}\left(\frac{x^2}{L} - x + C\right) \qquad (1)$$

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The maximum moment will again occur in the middle, *i. e.*, when $x = \frac{L}{2}$.

$$M_b = \frac{W+U}{8} L \qquad (3a)$$

Both maximum moments, Figs. 3 and 4, are equal, except that the moments are oppositely directed in rotation about A or B. It is obvious now that there is a certain value for C in which the negative moment in the middle of the beam equals the positive moments at the supports A and B. As the bending moments between A and B change from a negative to a positive value, there will be two points O and O, Fig. 2, where the moments $M_x = 0$. In order to find this value for C, we simply make (Fig. 2) the moment in the middle equal the bending moments at the supports A or B; so that we have the condition:

$$-M_m = M_c \qquad (4)$$

or

$$-\frac{W+U}{2}\left(\frac{L}{4} - \frac{L}{2} + C\right) = \frac{W+U}{2}\left(\frac{C^2}{L} - C + C\right) \qquad (4a)$$

which equation, reduced, gives us the value for

$$C = \left(\sqrt{\tfrac{1}{2}} - \tfrac{1}{2}\right) L = 0.207\, L \qquad (5)$$

The value substituted in equation (4a) gives for the maximum moment

$$-M_m = M_c = \frac{W+U}{47} L \qquad (6)$$

This formula indicates that a continuous beam of the length L, if supported symmetrically at the distance $C =$

$0.207\,L$ from the ends, will possess the greatest carrying capacity. As regards the methods of supporting in the cases in Fig. 3 and Fig. 4, the continuous beam will carry about 6 times more weight. This, naturally, applies also to cylindrical drums supported on rollers.

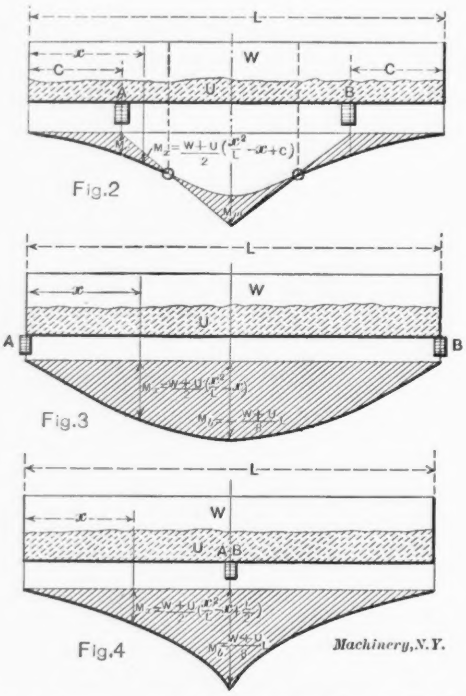
The moment of inertia of a circular ring section is

$$I = \frac{\pi}{64}(D^4 - d^4) \qquad (7)$$

and the section modulus

$$R = \frac{\pi}{32}\,\frac{D^4 - d^4}{D} \qquad (7a)$$

From formula (7a) by combining with the general equation



Figs. 2, 3 and 4. Bending Moments for Drum supported at Different Places.

(1) for the bending moment, we get for the maximum fiber stresses in bending:

$$S = \frac{16\,D\,(W + U)}{\pi\,(D^4 - d^4)}\left(\frac{x^2}{L} - x + C\right) \qquad (8)$$

As these drums are built up of several single shells, it will be a good plan to distribute the joints in such a manner so that no joints come at the places of maximum bending, that is in the middle and at the supporting points. Then, at the joints where bolt and rivet holes weaken the material, the above fiber stress will have to be divided by the efficiency of the joint, which, according to best practice, seldom exceeds 85 per cent. Care should also be taken to provide a sufficient number of rivets, to take care of all occurring shearing stresses, such as we have in common built-up girders. Finally, stiffeners will have to be provided at intervals around the drum to keep it in shape. In barrels, which have to be provided with openings, the decrease in strength may, to a certain extent, be compensated for by reinforcements, such as bars, angles and beams (Fig. 5).

As the drum is rotating, the maximum fiber stress will alternately assume postive and negative values, similar to those we have in connecting-rods. On that account, the maximum fiber stress should be lower than the allowable stress for bridges or girders. Then, too, the drum is not only subjected to stresses by flexure alone; besides the shearing stresses already mentioned, there is also a torsional shearing stress due to the rotating power, which produces a twisting moment M_t in the drum. We may combine the bending and twisting moments to an ideal moment, by means of the Reuleaux formula—

$$M_o = \tfrac{3}{8}\,M_b + \tfrac{5}{8}\sqrt{M_b^2 + M_t^2} \qquad (9)$$

From this the actual maximum fiber stress may be calculated by dividing formula (9) by the ideal section modulus R_o.

$$S_o = \frac{M_o}{R_o} \qquad (10)$$

$R_o = R \times$ coefficient of strength of joint.

The torsional moment is easily figured from the frictional resistance of the supporting rollers and the turning moment of the material inside the drum.

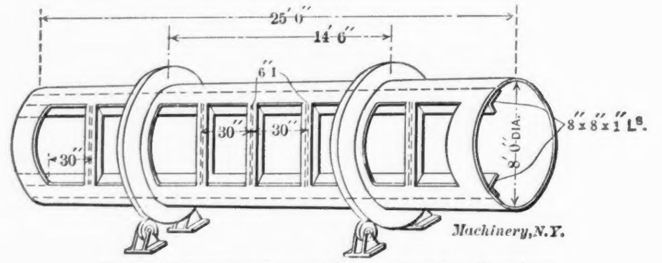


Fig. 5. Schematic View of Drum, showing Openings.

In order to illustrate this more clearly, we may proceed now with a specific example. Fig. 5 shows a drum used in a certain manufacturing operation, which is required, when it is filled, to carry a load $U = 200{,}000$ pounds of a material resembling sand, weighing, say, about 160 pounds per cubic foot. The natural slope of the materials is ~ 45 degrees when calcined, and the weight is then 90 pounds. On account of the bolt or rivet holes for the supporting rings, we make M_c 10 per cent less than M_m, and derive then C, according to formula (4a) as follows:

$$-\,0.9\left(\frac{L}{4} - \frac{L}{2} + C\right) = \frac{C^2}{L};$$

$$C = \tfrac{1}{2}\,(\sqrt{1.71} - 0.9)\,L = 0.204\,L.$$

As the length $L = 25$ feet, we have $C = 5.1$ feet.

Therefore the most advantageous distance between the supporting rollers is $L - 2C = 25 - 10.2 = 14.8$ feet $\cong$ 14 feet 6 inches.

This distance would produce the least bending moment if the load is distributed over the entire length of drum. While

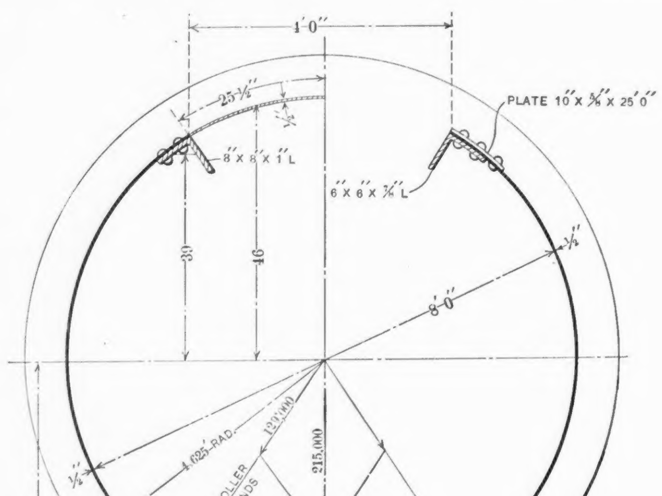


Fig. 6. Section through Drum.

in operation, however, such circumstances may probably occur that the load is mainly distributed between the supporting rollers, in which case it will throw a greater bending stress on the drum. Consequently, in the preliminary and final calculations for the required shell thickness, we will consider as the bending moment:

$$M_m = \frac{U\,(L - 2\,C)}{8} = \frac{200{,}000 \times 14.5}{8} = 362{,}500 \text{ foot pounds.}$$

To allow for sufficient stiffness, on account of the large diameter of the drum, the extreme fiber stress should be taken very low, say $S = 2{,}000$ pounds per square inch, giving the value for the moment of resistance

$$R = \frac{M_m}{S} = \frac{362{,}500}{2000} = 181.$$

[The moment of resistance is arrived at in the manner in common use with structural designers, lengths being expressed in feet, but stresses in pounds per square inch. In fact, the value for R then becomes merely a constant.—EDITOR.]

According to this value, we find in the accompanying table the nearest thickness of shell to be 5/16 inch. From the preliminary estimate we can now "guesstimate" that a thickness of ½ inch will be about right, when considering the additional torque, the efficiency of the rivet joints, and the abrasion inside the drum by the material, as well as the

SECTION MODULUS FOR VARIOUS THICKNESSES OF TUBES, DRUMS, STACKS AND STAND-PIPES.

For diameters in inches, length in feet, stresses in pounds per square inch, and moments in foot-pounds.

Inside Diam. in inches.	3/16	1/4	5/16	3/8	7/16	1/2	9/16	5/8	Inside Diam. in feet.	Weight per foot 1 inch Thick.
18	4.02	5.38	6.75	8.11	9.52	10.9	12.4	13.7	1′ 6″	202.9
20	4.96	6.63	8.32	9.99	11.7	13.4	15.3	16.9	1′ 8″	224.5
24	7.13	9.53	11.9	14.3	16.8	19.3	21.7	24.2	2′ 0″	267.0
28	9.69	13.0	16.2	19.5	22.8	26.1	29.5	32.8	2′ 4″	309.8
32	12.6	16.9	21.2	25.4	29.7	34.1	38.4	42.7	2′ 8″	352.5
36	16.0	21.4	26.8	32.1	37.6	43.0	48.5	53.9	3′ 0″	395.2
40	19.7	26.4	33.0	39.6	46.3	53.0	59.8	66.5	3′ 4″	438.0
44	23.9	31.9	39.9	47.9	56.1	64.1	72.2	80.4	3′ 8″	480.8
48	28.3	37.9	47.4	56.9	66.5	76.1	85.8	95.4	4′ 0″	523.5
52	33.4	44.5	56.7	66.8	78.1	89.4	101.0	112.0	4′ 4″	566.2
56	38.6	51.6	64.7	77.5	90.5	104.0	117.0	130.0	4 8″	609.0
60	44.6	59.3	74.0	89.0	104.0	119.0	134.0	149.0	5′ 0″	651.7
64	50.4	67.3	84.1	101.0	118.0	135.0	152.0	169.0	5′ 4″	694.4
68	56.8	75.9	94.9	114.0	133.0	152.0	172.0	191.0	5′ 8″	737.2
72	63.8	85.0	107.0	128.0	149.0	171.0	192.0	214.0	6′ 0″	779.8
78	74.8	100.0	125.0	150.0	175.0	200.0	226.0	251.0	6′ 6″	843.9
84	86.8	116.0	145.0	174.0	203.0	232.0	261.0	291.0	7′ 0″	908.0
90	100.0	133.0	166.0	200.0	233.0	267.0	300.0	334.0	7′ 6″	972.1
96	113.0	151.0	189.0	227.0	265.0	303.0	341.0	380.0	8′ 0″	1036.0
102	....	171.0	213.0	256.0	299.0	342.0	385.0	428.0	8′ 6″	1100.0
108	....	191.0	239.0	287.0	335.0	383.0	432.0	480.0	9′ 0″	1164.0
114	....	213.0	267.0	320.0	373.0	427.0	481.0	535.0	9′ 6″	1228.0
120	....	236.0	295.0	354.0	414.0	473.0	533.0	592.0	10′ 0″	1293.0
132	....	286.0	357.0	429.0	501.0	572.0	644.0	716.0	11′ 0″	1421.0
144	....	340.0	425.0	510.0	596.0	681.0	766.0	852.0	12′ 0″	1549.0
156	....	399.0	499.0	599.0	699.0	799.0	899.0	999.0	13′ 0″	1677.0
168	....	462.0	578.0	694.0	810.0	926.0	1043.0	1160.0	14′ 0″	1805.0
180	....	531.0	664.0	797.0	930.0	1064.0	1197.0	1331.0	15′ 0″	1934.0
192	....	604.0	755.0	906.0	1058.0	1209.0	1361.0	1513.0	16′ 0″	2062.0
204	....	682.0	853.0	1024.0	1194.0	1365.0	1536.0	1708.0	17′ 0″	2190.0
216	....	764.0	956.0	1147.0	1339.0	1530.0	1722.0	1914.0	18′ 0″	2318.0
228	....	852.0	1065.0	1278.0	1492.0	1705.0	1919.0	2133.0	19′ 0″	2446.0
240	....	943.0	1180.0	1416.0	1653.0	1889.0	2126.0	2362.0	20′ 0″	2574.0

For moments expressed in inch-pounds, multiply the corresponding value in table by 12.

considerable dead weight of the drum itself. This latter, according to the table amounts to $\frac{1036}{2} = 518$ pounds per running foot, not including the weight of joints and rivets, also necessary stiffening angles around the drum, etc. Assuming these to be 15 per cent of the weight, the total weight of drum amounts to about $W = 518 \times 1.15 \times 25 = 14892 =$ 15,000 pounds approximately.

If it is desired to find the amount of reinforcement which will exactly compensate for the cutting away of the plate for the openings of 4 feet width, it will be sufficiently accurate to make the momentum of plate section equal the momentum of reinforcing angle, so that (Fig. 6)

$$25\tfrac{1}{2} \times \tfrac{1}{2} \times 46 = A \times 39.$$

From this follows the required area of reinforcement

$$A = \frac{586.5}{39} = 15 \text{ square inches.}$$

Selecting the commercial size $8 \times 8 \times 1$ inch angle, of 15.25 square inches area, this allows also for the rivet holes. Taking, for instance, a $6 \times 6 \times \frac{7}{8}$-inch angle of 9.74 square inch area, an additional strap $10 \times \frac{5}{8}$ inch thick will be required, as shown to the right in Fig. 6.

We are now ready to figure the maximum torque required to rotate the drum. Referring to Fig. 6, if the sum of the weight of contained material and of the drum equals 215,000 pounds, then the pressure on each of four rollers will be 64,500 pounds. If the coefficient of friction is assumed to be 0.08 (which would be about right for a proportion of d to D as one to four), we have $0.08 \times 64{,}500 = 5{,}160$, and if the radius of the drum at its contact line with the rollers is 4,625 feet, we have for the torque:

$$4 \times 5{,}160 \times 4.625 = 95{,}460 \text{ foot-pounds.}$$

The weight of the shell produces the bending moment

$$-M'_m = \frac{W}{2}\left(C - \frac{L}{4}\right) = \frac{15{,}000}{2}(5.1 - 6.25),$$

$$M'_m = 8625 \text{ foot-pounds.}$$

Obviously, the total maximum bending moment in the drum

$$M_b = M_m + M'_m = 362{,}500 + 8625$$
$$= 370{,}825 \text{ foot-pounds.}$$

Then the ideal bending moment, formula (9)

$$M_o = \tfrac{3}{8} \times 370{,}825 + \tfrac{5}{8}\sqrt{370{,}825^2 + 95{,}460^2}$$
$$= 378{,}500 \text{ foot-pounds.}$$

Taking the strength of the rivet joint about 50 per cent of that of the full plate, it follows, from the table, the actual moment of resistance for a shell thickness of ½ inch:

$$R_o = \frac{303 \times 50}{100} = 151.5,$$

which finally gives us the corrected value for the maximum fiber stress, from bending and twisting,

$$S_o \frac{378{,}500}{151.5} = 2500 \text{ pounds per sq. in.}$$

There are varied opinions regarding the allowable stress as well as regarding the friction coefficients, and it is extremely important that the practical engineer should use good judgment and reliable data. Frequently, the material undergoes considerable changes before and after the process. The loss of strength through the higher temperature will have some influence on the thickness of the shell; and it will be almost impossible to give any more definite general rules to the designer for proportioning the drum and its parts than the outline of the process of calculation presented above.

By the continuous calcining process, Fig. 1, the torque will always remain approximately constant, while, as a rule, the intermittent working dryers, as here presented, will require variable power for turning the drum during an operation period. Not seldom the maximum torque is at a certain stage of the operation, and not at the beginning or end of the process.

[It will be noticed that in the above calculations reference has not been made to the influence on the twisting moment by the distribution of the material in the drum, which influence, of course, increases with speed of rotation, and depends on the number and location of cascade bars and the natural slope assumed by the material. The latter condition, of course, produces an eccentric loading, but for a given slope of material it is a comparatively easy matter to determine the center of gravity of the material, and thus figure the torque with reference to the central axis.—EDITOR.]

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An investigation of man-hole explosions at Aberdeen, Scotland, discloses the fact, says the *Engineering Record*, that coal gas, leaking from street mains, may become odorless by filtering through a moderately thick layer of earth without losing its explosive effect.

FOUR 4000 H. P. GAS ENGINES.*

JAMES COOKE MILLS †

In the rebuilding of San Francisco from the very bed-rock, one might say, advantage has been taken of new discoveries, mechanical devices and inventions to provide the wheels of industry with the best power-producing machinery yet known to man. This spirit of substantial progress is no more evident than in the construction and installation of the most modern power plant in this country, for the California Gas and Electric Corporation, furnishing current for all the railways controlled by the United Railways.

The new electric plant has been in constant service for several months, auxiliary to the system which is fed by hydro-electric plants and two steam plants; and the success of the gas engines is so marked that it is the intention of the public service corporation to install more of the gas engine units until, in all probability, the output will amount to 50,000 H. P.

The plant now comprises four units, each of twin-tandem, double-acting gas engines, operating on crude oil water gas and rating at 4,000 B. H. P., with one-third overload capacity momentarily, or 4,600 B. H. P., on the usual overload rating of 15 per cent, at which rating the engines have frequently run for several days without inconvenience or interruption in the service. They are by far the largest and most powerful gas engines ever constructed, and the illustrations, Figs. 1 and 2, give some idea of the massive steel frames, huge cylinders, and the complicated valve sets and gears. Between the twin engines, and driven by the main shaft, is the Crocker-Wheeler 25-cycle alternating-current generator. One gas engine unit of the same size drives a 60-cycle A. C. generator.

The cylinders are cast in two parts, the joints being circumferential and placed half way between the ends, and coupled up by broad flanges. They are supported on pedestals and rigidly bolted to the tie piece between them carrying the guides for the piston rod, on top and bottom. The front end of the forward cylinder bolts direct to the frame block, and this, in turn, is bolted to the foundation by long bolts extending clear through the frame to the top. This design gives the full height and strength of the frame to insure against vibration. The frame block carries the cross-head with guides on top and bottom, and also the main shaft block for the two main journals, which are 30 inches diameter and 56 inches long. The fly-wheel is 23 feet in diameter and weighs 97,000 pounds for the 25-cycle units, and 135,000 pounds for the 60-cycle units. The main blocks weigh 93 tons each, and the total weight of the engine is 600 tons. The main shaft alone weighs 52 tons, and the cranks, pins, piston-rods and all working parts are of unusual size and strength. With four double-acting cylinders, each 42 inches diameter and of 60-inch stroke, the engine is compactly built, and the space amounts to 74 by 35 feet—a space comparatively small for a gas engine of this type. The speeds of the engines are 88 and 90 revolutions a minute for the 25- and 60-cycle units respectively.

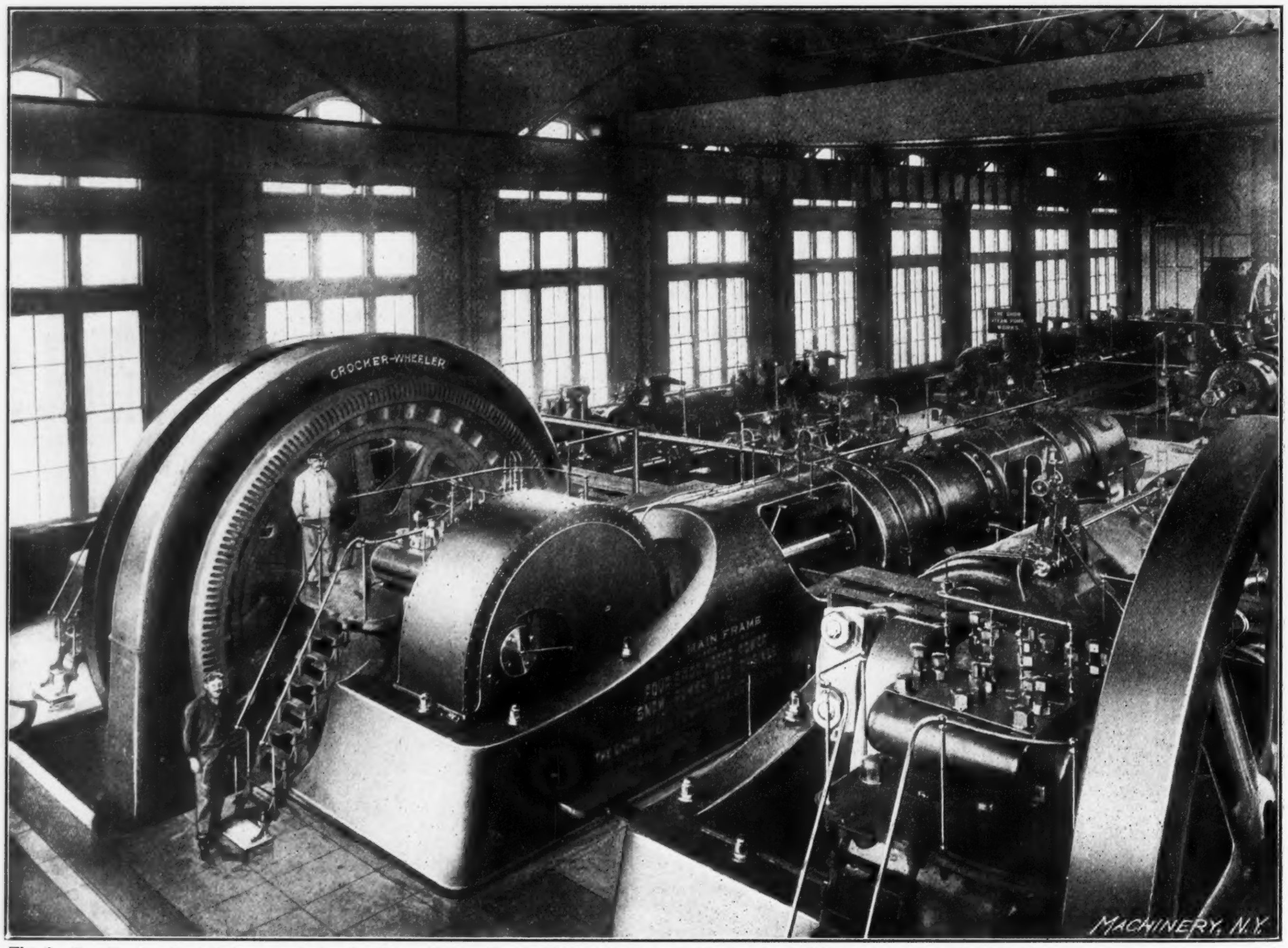

Fig. 1. Gas Engine Installation made by the Snow Steam Pump Works, Buffalo, N. Y., for the California Gas and Electric Corporation, San Francisco, Cal.

The tail-rod slides in guides, shown in the illustrations, and relieves the cylinder walls of wear. It adds a third support for the piston-rod, the cross-heads in front of the forward cylinder and between the cylinders in the tie piece providing the other two. The piston-rod is 15 inches diameter, the crank-pin is 19 inches diameter and of the same length. The wrist-pin is slightly smaller, being 17 inches diameter and 18 inches long. These figures give an idea of the massive construction employed throughout.

The valves are all located on the same side of the cylinders, the intake valves being above and the exhaust valves below. They are operated from a common lay-shaft geared to the main shaft. Their action is positive, being operated by vertical rods, rocker-arms, and cams. The inlet valve, mixing chamber, and cut-offs are designed so that gasoline can be injected to the surfaces which require cleaning, and any deposit is removed without touching the parts.

All cylinder working parts—cylinders, pistons, and piston

* For additional information on this and kindred subjects, see the following articles previously published in MACHINERY: Gas vs. Steam, December, 1901; The Vogt Gas Engine, February, 1904; Formulas and Constants for Gas Engine Design, February, 1906; Gas Engine Economy, May, 1906; A Naval Gas Engine, July, 1906; Alcohol as a Fuel for Gas Engines, August, 1906; Gas Turbine, March, 1907.

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rods—are thoroughly water-jacketed, each part being supplied with a separate water pipe, so that the amount of water may be regulated. In this way the cylinder may carry a high temperature, the cylinder jackets a medium temperature, and the rods and metallic packing a low temperature.

Lubrication is effected by means of an individual oil-pump with four leads to each cylinder. The oil is taken in on the inhalation stroke, spread on the compression stroke, and made ready for the working stroke which follows next after in the order of the cycle. Positive feed lubrication for the journals is provided by pipes leading from a multiple feed oiler.

The crude oil water gas supplied to these engines is generated by the Lowe system. It is a notable fact that the engines can be started and synchronized to 88 revolutions a minute, with hydro-electric and steam plants, in from 55 seconds to two minutes from the moment of receiving the signal. This includes turning on the compressed air for the starting, the gas, oil, electric current for the igniters, and coming up to full service speed.

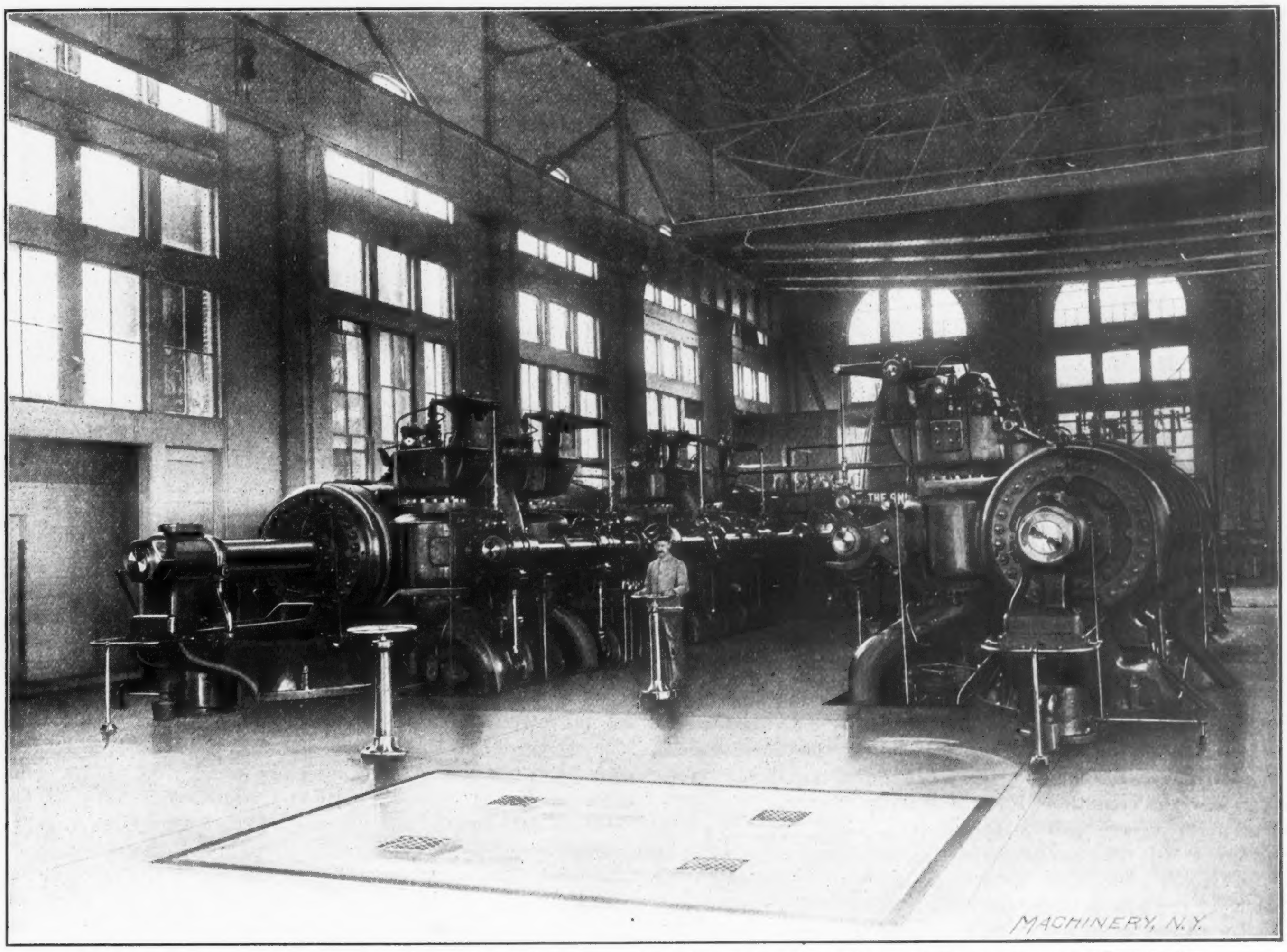

Fig. 2. View of a 4000 B. H. P. Gas Engine Unit from Floor Level. Note Tail-rod Supports and General Massiveness of the Construction.

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VANADIUM AND AUTO CYLINDERS.

One of the most serious features of deterioration in cars is the loss of compression through the wearing of the cylinders. Foreign engine castings always have been superior to any American products in this respect, but few have understood why. Light has been thrown on the subject recently by some experiments made by the American Locomotive Automobile Company. It was found that the inside of the cylinder castings obtained here will take a polish from the piston action quite readily, but that very soon after the inside polish has reached its best, it begins to check and crack away, leaving roughness. To this is due the loss of compression. Some imported Berliet cylinders, tried under the same conditions, took higher polish and held it, the compression showing no loss after long service. The fact that the Berliet castings have a considerable percentage of vanadium is believed to partly explain the difference.—*Graphite.*

THE VARIATION OF THE STRENGTH OF GEAR TEETH WITH THE VELOCITY.*

RALPH E. FLANDERS.†

The generally accepted formula for calculating the strength of gear teeth is that proposed by Mr. Wilfred Lewis, first published in the Proceedings of the Engineers' Club of Philadelphia, January, 1893.

The merit of this formula lies in the great number of variables taken into account as compared with other rules in more or less common use, and in the fact that these variables are rationally considered. The effect of each of them can be calculated with some assurance, with the single exception of the influence of the velocity on the safe stress. In the fourteen years since the formula was first proposed, the original values for the stress as affected by the velocity have been largely used. Many designers, however, have felt that these values are rather unsatisfactory, although most of them will agree that they err rather on the side of safety than otherwise. By referring to Mr. Lewis' original paper it will be seen that these values were not given as being definitely determined, but merely as agreeing well with successful cases met with in his own practice. It has recently been proposed, by some individuals interested in the matter, to undertake a series of tests for determining this as yet undetermined factor. This article is written with the purpose of calling attention to this matter, with the hope of bringing out suggestions as to the conduct of such tests; the writer also desires to indicate some considerations which he believes should be taken into account in making the experiments.

* For additional information on this and kindred subjects, see the following articles previously published in MACHINERY: The Pitch and Strength of Gear Teeth, March, 1895; The Strength of Gear Teeth, October, 1895; Proportion of Gear Teeth, September, 1896; Strength of Gear Teeth, September, 1896; Strength of Gear Teeth, February, 1897; Diagrams for Relative Strength of Gear Teeth, July, 1897; Formulas for the Strength of Gear Teeth, October and November, 1898; Gearing—2, Calculations for Strength and Power Transmitted, March, 1902; The Strength of Gear Teeth, engineering edition, October, 1906; Strength of Gears, engineering edition, December, 1906. See also MACHINERY'S data sheets: July, 1903; October, 1903; February, 1905; October, 1906; December, 1906.

† Associate Editor of MACHINERY.

Variation in Strength is Due to Impact.

A variation in the strength of the teeth of a gear, due to a variation in the velocity, can be due, of course, to but one thing—impact. To illustrate this idea, and to show the cause of the impact, we will study the action of gearing under three different conditions. First, when made of an imaginary material which does not deflect under any strain below the breaking point. Second, with gears of commercial material, such as steel, with teeth of perfect form. Third, gears of commercial material with teeth of commercial accuracy.

1. *Gears of an imaginary undeflectable material.*—In Fig. 1 is a diagram in which the horizontal distances give velocity in feet per minute, and vertical distances give stresses in pounds per square inch, starting in this case at 4,000, which is assumed to be the maximum fiber stress in the gear we are considering, due to the load at the pitch line, which is supposed to be constant at all speeds. If the teeth of this gear are perfectly formed and well fitted together, so that there is no back lash, if the power is delivered to them

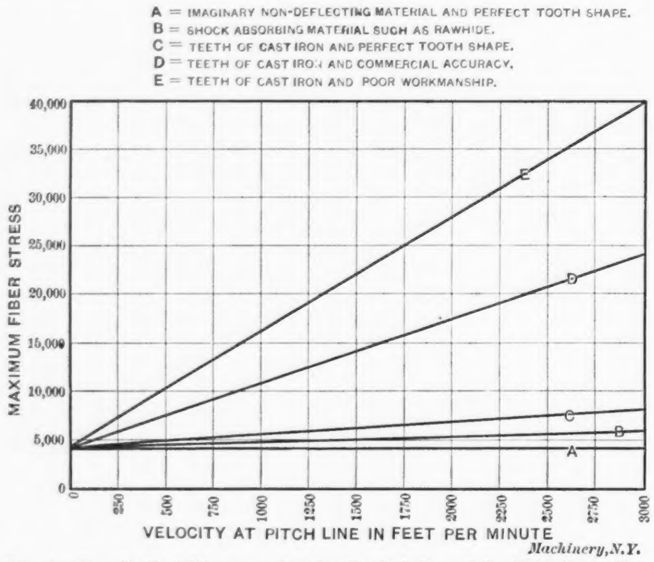


Fig. 1. Hypothetical Diagram showing the Relation of the Velocity to the Fiber Stress.

steadily and smoothly, and the mechanism they drive runs without shock, any disturbance of the even movement will be impossible, and impact will be entirely absent. In the diagram in Fig. 1, then, there will be no rise of maximum fiber stresses with the velocity, so that the horizontal line *A* will show the conditions for this imaginary case.

2. *With commercial material and theoretically accurate workmanship.* The conditions in this case are shown in Fig. 2, with all the phenomena greatly exaggerated. The full lines show the conditions under load, while the dotted outlines show the conditions when the load is removed from the driven gear. The teeth A_1, B_1 and A_2, B_2, carrying the load, are deflected by it, as shown. Tooth *B*, just about to come into contact with tooth *A*, is on that account shifted from its normal position; it should be located as shown by the dotted lines. If it were in this position, it would come in contact with tooth *A* under mathematically perfect conditions, and there would be no shock of engagement. As it is, the two come suddenly into action as shown at *E*, under different conditions than those contemplated by the design, thus the contact takes place in the form of a slight blow, after which the teeth are deflected more and more, until they have taken up their share of the load, as shown later at A_1 and B_1. If the gears are moving very slowly, the deflection takes place very slowly, and the problem is practically a static one. If the gears are running at a high velocity, the problem becomes essentially a dynamic one, and the stresses induced are greater than with the slow speed.

The increase in stress with the increase in speed for this second case could probably be represented by a line something like *C*, of Fig. 1. The location of this line is purely hypothetical. All we can say about it is that the increase in stress as the speed is increased would be comparatively small, and probably regular. The line has been drawn straight for convenience; we do not know what the real shape is.

3. *With commercial materials and commercial accuracy.* This is, of course, the practical case to consider. A line to show the relation of the velocity to the maximum fiber stress for a given gear, would very probably look something like *D* in Fig. 1. This is, in fact, approximately the line which embodies the conclusions of the Lewis tables for a static stress of 4,000 pounds. It is considerably higher than line *C*, because impact due to irregular tooth outlines is added to the impact due to the deflection. In all probability the latter is comparatively unimportant as compared to that due to irregularity of outline in gears of only ordinary workmanship.

Deflection Due to Impact is Slight, but Stresses are Great.

It may be objected that the deflections produced either by the gears coming into mesh out of step, as in case No. 2, or with the added aggravation of poor workmanship, as in case No. 3, are so minute that they could scarcely be considered as a serious factor in the problem. It is true that these deflections are minute—undetectable even, by ordinary means; but this admission does not destroy the argument for laying to this distortion the increase of the stress with the speed. If great loads produce slight deflections, slight deflections likewise produce great stresses, so that the slight bending brought about by the teeth coming into contact at *E* in Fig. 2, under slightly imperfect conditions, may produce great effects proportionately in the fiber stress, and the effects are magnified by the irregularities due to poor workmanship. When we stop to figure out what a load per inch of face is required to deflect a 2-inch circular pitch gear, say 0.003 inch, it is evident that an irregularity in outline of this amount would scarcely be negligible at high speeds, if our hypothesis is correct.

The phenomena of impact are complicated to a high degree. The maximum stresses produced depend on the rapidity of transmission of a wave of stress or deflection produced in the material by the impact. If this wave is propagated slowly, the stresses are nigh; if rapidly, the stresses are low. The factors entering into the problem are the elasticity of the material, and the mass and shape of the part affected. In very simple cases the problem has been investigated mathematically, but our problem with the gear teeth is so complicated that we must of necessity at once apply to the engineer's court of last resort—experiment.

Practical Considerations Affecting the Conduct of the Proposed Tests.

It is now evident that other variables besides the strength of the material and the velocity at the pitch line enter into the fixing of the line on the diagram of Fig. 1. In addition, the following points will have to be considered.

1. *Accuracy of tooth outlines.* From what has just been said, it is evident that the variation of the stress with the velocity will be affected by the accuracy of the workmanship involved in forming the tooth of the gear. Investigating the conditions in the case of a second pair of gears, similar to those from which line *D* was determined, but of a considerably poorer grade of workmanship, we should expect to find results giving a line something like *E* on the same diagram, giving much higher values for the stresses resulting from the load. It is evident, then, in considering lines *C*, *D* and *E* that workmanship is a variable which should be considered in the experiments, and that a series of tests should be run with two sets of gears of varying workmanship, one of high and the other of only ordinary grade, to make sure that this consideration is really of importance.

2. *Design of wheel and mechanism.*—Another factor which may affect the increase of the stress with the speed is the design of the rim and spokes of the wheel. It is conceivable that a gear with a very heavy rim and rigid spokes will absorb the shocks due to high velocity less easily than a gear with a light rim and flexible spokes or arms. The whole structure of the machine in which the gearing is carried, so far as its rigidity and massiveness is concerned, should, in fact, affect this matter. The further away from the point of tooth contact the members of the structure are, however, the less effect will they have, so perhaps even the influence of the arms and rim can be neglected. The same consideration affects the design of the mechanism to be used in the tests. It is conceivable that a mechanism involving long shafts and other flexible members

might give, for a given set of gears, a line lower down on the diagram of Fig. 1 than would be the case if the construction were very heavy and rigid. The supporting mechanism must not in any case, of course, deflect in such a way as to prevent the teeth from having a full bearing on each other.

3. *The nature of the materials used.* Referring to what has previously been said as to the factors governing impact, it will be seen that the nature of the material used would affect the shape of the curve. It is probable, for instance, that two sets of gears, one made of cast iron and the other of a bronze alloy of the same tensile strength, would show lines of very different shape, owing to the difference in the modulus of elasticity and the specific weight of these two substances. From this it will be seen that we cannot be sure that the results found to be applicable to cast iron or steel could also be applicable to either a pair of bronze gears or to the case of a bronze gear meshing with a mate of steel or iron. That the nature of the material would have a vital effect on the shape of the curve is still more probable, when we consider the practice followed in the use of such substances as rawhide. This material is particularly fitted to sustain impact and absorb without undue stress the deflection caused by it. Owing to this characteristic, we might expect that the line for a gear of this substance would be practically horizontal, as shown at *B* in the diagram, approaching *A*, though governed by entirely different conditions from those producing *A*. So far as can be learned, this supposition agrees with the practice of the manufacturers of rawhide gears.

It may be found that the points mentioned have so much influence on the question at issue that it would be very difficult to lay down a law governing the variation of stress with velocity, and that the most that can be done is to determine the variation in cases of commercial workmanship and rigid design, using the relation thus established in an empirical formula, with the knowledge that poorer conditions may bring the fiber stresses much higher, while good workmanship and careful design may, on the other hand, bring them much lower. Qu e possibly the factors now in use may be found to nearly fill commercial requirements, in which case we must conclude that the criticisms of their being too high have been founded on experience with cases combining the favorable conditions just mentioned.

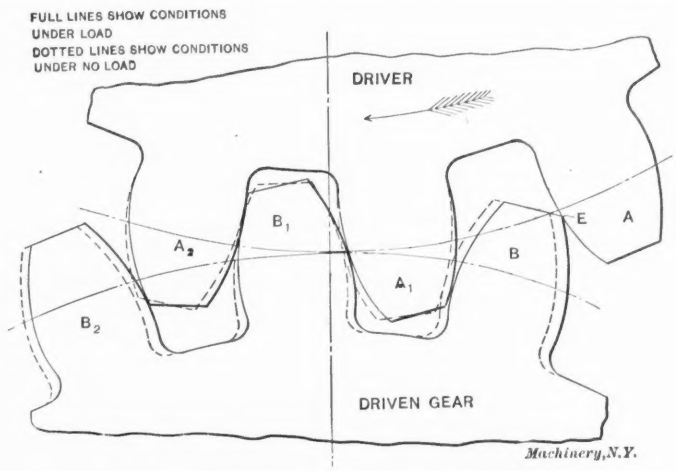


Fig. 2. The Action of Gear Teeth under Load, Greatly Exaggerated.

Practical Considerations Affecting Design.

The fact that the variation of the strength with the velocity is due to impact, suggests also a number of points relating to design. Most of these are already well known, and are standard practice, the conclusions being so obvious that simple common sense has suggested them without theoretical analysis being necessary.

1. *Value of accuracy.* It is evident that this theory of impact puts a premium on accuracy in workmanship for gears that are to run at high speed under a heavy load. It is probable that the strength of a given pair of gears may be cut in two if the tooth outlines are not carefully determined, and if the cutter is not set centrally. This suggests the desirability of a greater sub-division of the standard cutter series for work of this kind. Of course, the gears can always be made heavy enough for the required service, but the extra cost of accurate cutters and careful cutting will be repaid in cases where light weight and compact design are at a premium. In such cases the use of cutters specially designed for each gear is recommended.

2. *Resilience of design and materials.* In high-speed gearing it is evident that the shock due to the impact should be absorbed as quickly and as fully as possible. This suggests the use at abnormally high speeds of rawhide, wood, etc., for one of the members of the pair of gears. The introduction of spring couplings or similar devices may also be desirable, especially where the other parts of the mechanism are liable to transmit shock to the gearing.

3. *Easing off the points of the tooth.* As was suggested to the writer a short time ago by Mr. Fellows, of the Fellows Gear Shaper Co., this matter of impact affects gear tooth design in another way. There has always been a sort of superstition that the points of the tooth should be eased off to make the action smoother. This is done, of course, in standard involute gears, though for another reason, that of avoiding interference with the flanks of the pinions. It can now be seen that there is a solid basis for this practice in all cases where gears are to run at such speeds that severe impact is liable to take place. Referring to Fig. 2, teeth *A* and *B* are taking up the load very suddenly, owing to the fact that they are out of step, due to the deflection of the other teeth momentarily carrying the load. Easing away the points of *A* and *B* would mitigate this sudden reception of the load, allowing the inevitable deflection to take place more slowly, with a consequent gain in the strength of the gear at high speeds. It would have a similar effect in minimizing impact due to inaccuracy of outline.

This modification of the outline of the tooth should be very slight, and extend but a short distance, so that, when the load is entirely transferred, the "doctored" portion of the curve will be passed, and the true involute or cycloidal portion begun.

* * *

The question of aerial navigation is coming so much to the front that, while it is as yet only in its first experimental stage, any engineering review would be incomplete without a reference to what has been done in this territory of human achievement. Various European governments have already experimented with steerable airships for military purposes, and our own government has made an appropriation for the same purpose. A German steerable airship, according to a recent statement in the *Berliner Tageblatt*, was constructed in perfect secrecy, and has performed, with satisfactory results, a first four-hour trial run. The airship navigated at a height of about one mile with a speed of from 28 to 31 miles an hour, and showed a remarkable stability. The platform under the supporting balloon, which latter is of the spindle shape, affords accommodation for six persons and can be armed by automatic guns. Remarkable feats have also been accomplished by ordinary balloons. The results obtained from the international contest at St. Louis are well-known, but even more interesting feats have been accomplished on the other side of the ocean. The *Daily Graphic* of London financed a balloon expedition which recently left London with the hope of securing the world's record for long distance balloon traveling, which consists of a travel of 1,193 miles, from Paris to Russia. The *Daily Graphic* balloon did not accomplish this object, but its passage over the sea is of interest as being the longest over-sea balloon voyage undertaken. The balloon crossed the North Sea, then moved across Northern Denmark and over the Cattegat, until the descent was finally made in Sweden after a distance of about 1,000 miles entirely over the sea had been covered. The balloon, it is stated, was the largest that has ever been manufactured, having a capacity of 108,000 cubic feet. The car was so constructed that if it dropped into the sea, it would be capable of keeping afloat.

* * *

'Tis a shifting belt that has no crowned pulley.
A belt-shifter that sticks, and the man who does not as he is told shall be "cussed."

DECEMBER MEETING OF THE A. S. M. E.

The New York meeting of the American Society of Mechanical Engineers, held in December of each year, was a notable event this year, inasmuch as it was the first general meeting of the society held in the new Engineering Societies Building. This magnificent structure was dedicated last April 16 and 17 (see the May, 1907, issue), but the spring meeting of the A. S. M. E. always being held outside of New York City had prevented many of the members seeing the new building until the regular December meeting. That its spacious assembly rooms, splendid auditorium, fine library and many other carefully planned features were generally appreciated, there can be no doubt. Many were the expressions of pleasure in regard to the new society house and the manner in which the meetings were carried on.

Following is a list of papers presented and their authors:

The Mechanical Engineers and the Function of the Engineering Society (presidential address)..Prof. F. R. Hutton
The Rational Utilization of Low-Grade Fuels in Gas Producers ....F. E. Junge
Duty Test on Gas Power Plant..................J. R. Bibbins
Control of Internal Combustion in Gas Engines, Prof. C. E. Lucke
Evolution of the Internal Combustion Engine....S. A. Reeve
Industrial Education...........................W. B. Russell
The Foundry Department and the Department of Engineering Design..............................W. A. Bole
Molding Sand.............................A. E. Outerbridge
Power Service in the Foundry................A. D. Williams
Foundry for Bench Work......W. J. Keep and Emmet Dwyer
A Volumetric Study of Cast Iron................H. M. Lane
Specifications for Iron, Coke, and Method of Testing Output ....Dr. R. Moldenke
Foundry Cupola and Iron Mixtures...............W. J. Keep
Foundry Blower Practice........................W. B. Snow
Patterns for Repetition Work....................E. H. Berry
Some Limitations of Molding Machines.........E. H. Mumford
The Specific Heat of Superheated Steam...Prof. C. C. Thomas
Engine Design Adapted for the Use of Superheated Steam..................................Max E. R. Toltz
Power Transmission by Friction Driving..Prof. W. F. M. Goss
Cylinder Port Velocities..................Prof. J. H. Wallace

The fact that both the main auditorium and smaller assembly rooms were available for the meetings of the society permitted the papers to be discussed simultaneously. The papers were read in the main auditorium and the discussions of some, which were very lengthy, continued during the following sessions in the smaller assembly rooms, thus permitting full discussion by all who wished to take part.

On Wednesday afternoon between 300 and 400 of the members and guests crossed the North River on the Lackawanna ferry to the Lackawanna Railroad terminal station, and there met Mr. Chas. M. Jacobs, chief engineer of the Hudson Companies, who escorted the party on a tour of inspection through the tunnel. The route followed the North River shore from the Lackawanna Railroad terminal to 15th St., Jersey City, where the tunnel first built crosses the river. The party emerged from the tunnel at Morton and Hudson Sts., New York. The tunnel trip was a novel and interesting experience to the engineers and their friends, and nearly all thought it well worth the fatigue incident to the rough walking. It is expected that trains will be running in a few weeks.

The symposiums on foundry work and power development included the bulk of the papers presented, the papers by Mr. Russell: Industrial Education, and by Prof. Goss: Power Transmission by Friction Driving, being the exceptions. Mr. Russell's paper provoked considerable discussion, which is an indication of the interest in this present vital subject.

This meeting was the first in recent years in which all functions incident to the meeting, except excursions, were held in the Society house, including the reception and ball. This latter event for several years heretofore has been held at Sherry's.

There was a large registration of members and guests, the total number of members registered being 699; and of guests, 613.

The following officers were elected: President, M. L. Holman; vice-presidents, L. P. Breckenridge, Arthur West, Fred J. Miller; managers, Wm. L. Abbott, Henry G. Stott, Alexander C. Humphreys; treasurer, Wm. H. Wiley.

M. L. HOLMAN, PRESIDENT A. S. M. E.

Minard LaFever Holman, the newly elected president of the American Society of Mechanical Engineers, was born in Maine in 1852. His father was Col. Holman of the United States supervising architect's department. Col. Holman moved his family from Maine to St. Louis in 1860, and the subject of this sketch has lived in that city ever since. He graduated from Washington University of St. Louis in 1874 with the degree of Civil Engineer, and the University has since conferred on him the honorary degree of Master of Arts. For some time after his graduation Mr. Holman was connected with the supervising architect's department. He left that position to connect himself with the St. Louis Water Department under the late Thomas J. Whitman. After serving several years as principal assistant engineer under Mr. Whitman,

M. L. Holman.

he was appointed Water Commissioner of St. Louis in 1887. This office he held for three terms of four years each, making twelve years in all. Under his efficient supervision the St. Louis water works was almost entirely rebuilt, and experiments were inaugurated which resulted in a satisfactory clarification of the Mississippi River water, making it available for general city use.

After leaving the water department, Mr. Holman was for over four years the general superintendent of the Missouri Edison Electric Co., and for the last four years he has devoted his entire time to consulting work, having formed a partnership for this purpose with Mr. John A. Laird, member of the American Society of Mechanical Engineers, under the firm name of Holman & Laird. Mr. Holman has made a specialty of water works and power plants and has served on some important commissions, notably the one for designing new water works for Omaha. He was chairman of that committee, and he is now a member of the committee for appraising the property of the Denver Water Co., representing the city of Denver on the commission. Besides being a member of the A. S. M. E., Mr. Holman is a member of the American Society of Civil Engineers, American Institute of Electrical Engineers, St. Louis Engineers' Club, St. Louis Academy of Science, and is an honorary member of the American Water Works Association.

* * *

A contributor to the *Practical Engineer* states that it is possible to harden copper on its surface. This is accomplished by heating the copper, which is filed up clean before the process, to a bright red heat throughout its mass, and then plunging it into a pot of pure melted tin. The copper should remain in the tin until its heat is the same as that of the melted tin. The tin should be well covered with ground charcoal to prevent oxidation to any great extent. Proceeding in this way, the surface of the copper will be even harder than that of unhardened tool steel, and the hardness will penetrate to some depth. The tin alloys with the surface of the copper to cause hardness.

GEAR-CUTTING MACHINERY.—1.

RALPH E. FLANDERS.*

METHODS OF CUTTING GEAR TEETH.

There is no form of machine tool which has called for more ingenuity in design than the gear-cutting machine. The methods by which gears may be cut are so numerous, the requirements are so varied, the possible application of ingenious geometrical principles through the mechanism used are so nearly limitless, that a wonderful variety in design and construction has been evolved, affording a field of study which is unparalleled in its interest to the machine designer.

The earliest form of gear-cutting machinery to attain anything like its present state of development was the automatic spur gear machine using a milled cutter to shape the tooth. Later, came a period in which various forms of bevel gear cutting machinery were evolved, the demand being stimulated by the necessities of the chainless bicycle business. More recently, the requirements of the automobile have resulted in another period of inventivé activity, which has resulted in the development of new machines and processes for gears of all kinds, though the bulk of the attention has been given to the spur and bevel forms.

In the following pages we have attempted to cover the whole field of gear cutting, representing, so far as possible, every machine and process which has come into commercial use, including at the same time some which have been built and used successfully, but which have not been, for one reason or another, placed on the market. Some of these machines are old, some of them new, some of them simple, some of them complicated. Some of them are used for the finest kind of work, such as is required in watch and instrument gearing, while one of them will cut teeth in a gear up to 40 feet in diameter. The reader will see, as we proceed, that all these widely varying tools may be brought into a definite classification which links them all into one large family—the old and new, simple and complicated, large and small—by characteristics which are common to the different groups.

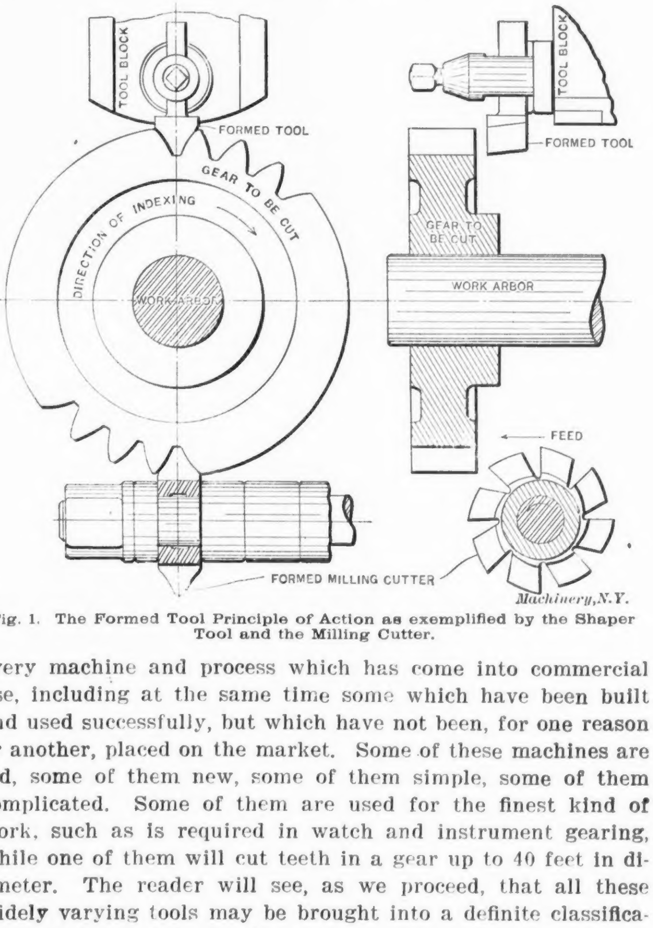


Fig. 1. The Formed Tool Principle of Action as exemplified by the Shaper Tool and the Milling Cutter.

The Classification of Gear-cutting Machinery.

Gear-cutting machinery may be classified, first, according to its *product*. There are four main divisions in this classification, separating from each other the machines designed for cutting spur, spiral, bevel and worm gearing, respectively. The cutting of internal gears and racks is analogous to the cutting of spur gears, and is included with it. Twisted or herringbone gears having parallel axes are in general cut in the same way as spiral gears, though, as gears, they belong to a different class. Some machines are so designed as to be capable of cutting more than one form of gear, but it is only done by making certain adjustments or using certain attachments which, for the time being, convert them into machines of other types. The best example of a machine which covers all the divisions of this classification is the universal miller, which may be arranged to cut the teeth in any one of the four forms mentioned.

The second classification of gear-cutting machinery depends on the *principle of action* involved. The five methods we will consider are—the formed tool, templet, odontographic, describing-generating, and molding-generating methods. This classification relates particularly to the way in which the tool is held and guided with reference to the work, to produce the desired form for the tooth surfaces.

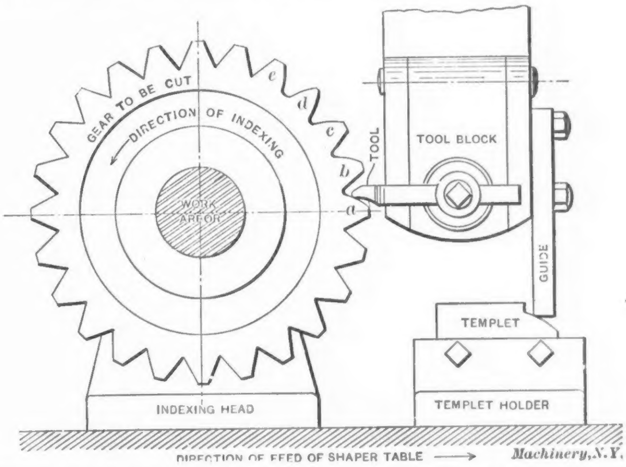


Fig. 2. The Templet Principle, as arranged to be applied to the Shaper.

The third method of classification relates to the *nature of the operation*. The four operations we will consider are—forming the tooth by impression, by planing or shaping, by milling, and by grinding or abrasion.

In studying the various combinations possible in these three different classifications, it will be simplest to first consider the matter of cutting the teeth of spur gearing, investigating the principle of action involved, and the nature of the operation performed, in the different methods. From that we will be able to proceed to the application of these principles and operations to the other forms of gearing, like the spur, bevel and worm.

Five Principles of Action.

The Formed Tool Principle: This, the simplest and most obvious way of forming a gear tooth, is illustrated in Fig. 1. The gear to be cut is held firmly on a work arbor which, in turn, is firmly supported in the machine, in such a way that it can be *indexed* (or rotated through an angular distance corresponding to one tooth) from time to time as occasion requires. In the upper part of the cut is shown a planer or shaper toolpost, carrying a formed tool having outlines accurately corresponding to the shape of a space between two of the teeth it is desired to form. It is evident that this formed tool,

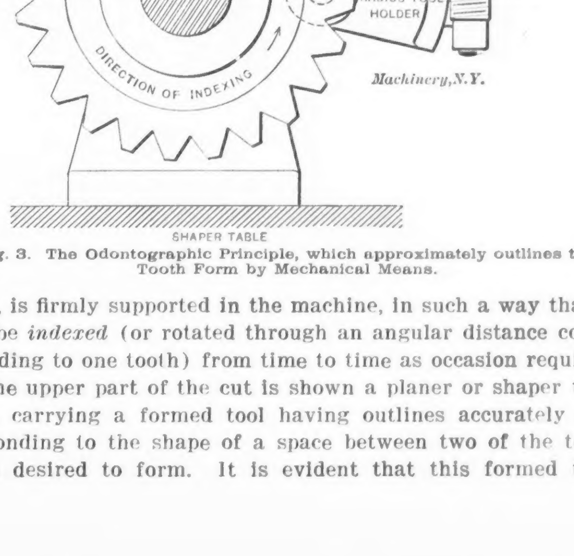


Fig. 3. The Odontographic Principle, which approximately outlines the Tooth Form by Mechanical Means.

* Associate Editor of MACHINERY.

when mounted in the tool-post of the planer or shaper, may be fed down into the work to the proper depth, in which case, being set centrally, it will reproduce its outline in the work. The work may then be indexed, and the operation repeated to form another tooth space. With the work indexed in the direction shown in the cut, four tooth spaces, or three complete teeth have been formed. A formed milling cutter may be used instead of the planer or shaper tool. This is shown

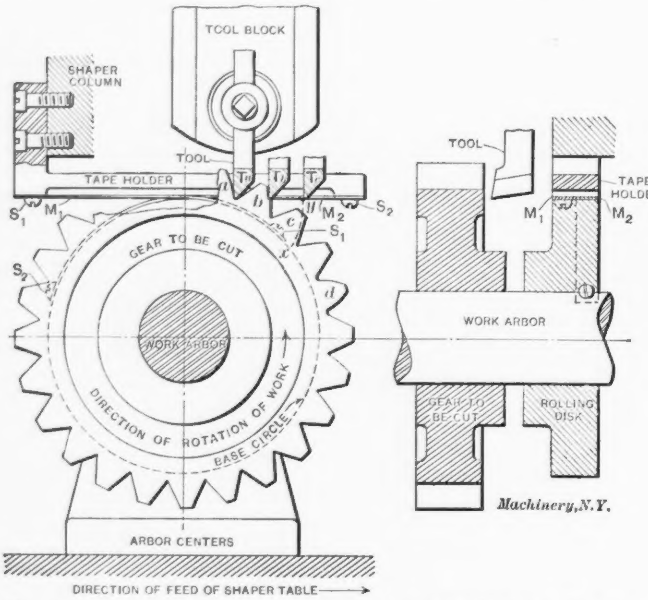


Fig. 4. The Describing-Generating Process, by which the Point of the Tool is constrained to follow the Desired Outline.

at work on the under side of the blank. It reproduces its outline in the work in the same way as does the planer tool, being rotated in the direction indicated, and fed through the work at the same time.

The Templet Principle: This method of cutting gears is shown in Fig. 2. As in the previous case, the work is held on the table of the shaper. A templet holder is also mounted on the shaper table, carrying a templet, having a surface formed to the exact outline desired for the finished tooth. The tool block is disconnected from the feed screw, and weighted

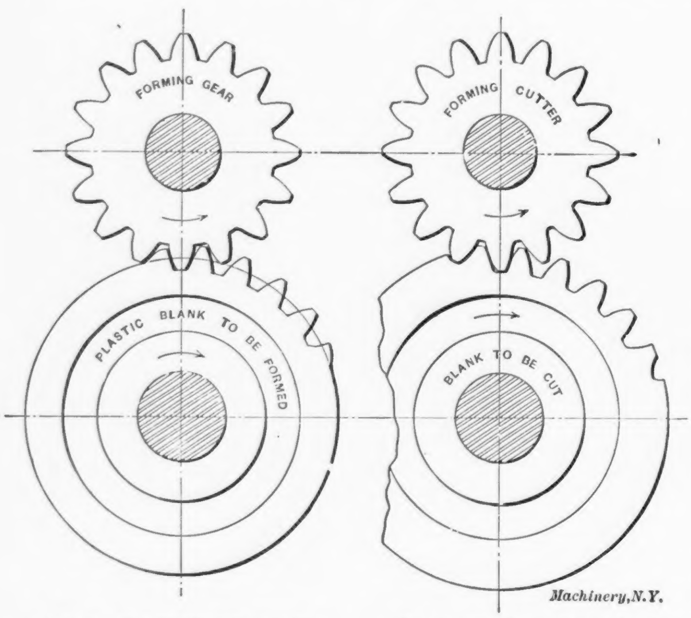
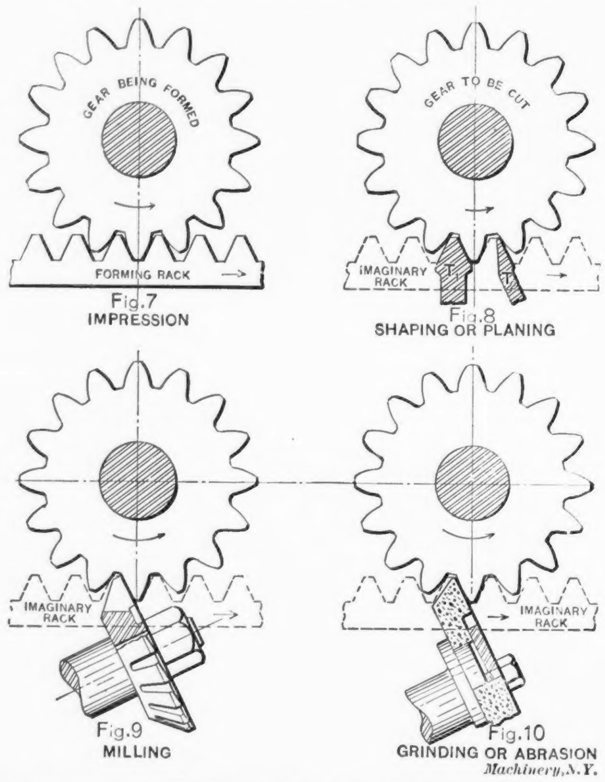


Fig. 5. The Molding-Generating Principle applied to Rolling the Proper Form in a Plastic Blank.

Fig. 6. The Same Principle employing a Cutter having a Shaping Action, cutting Teeth in a Solid Blank.

so that it falls of its own accord. To its side is clamped the guide shown, which bears on the templet. As the table of the shaper is fed to the right, it will be seen that the curved surface of the templet will raise the guide, the tool block and the tool, in such a fashion that the desired outline will be reproduced on the gear tooth. The upper surfaces of teeth *a*, *b*, *c* and *d* have been formed in turn in this way, the work being indexed for this purpose as in the previous case. With the primitive arrangement shown, it will be necessary to reverse the work in the arbor to form the other side of the teeth. Teeth *d* and *e* had their faces finished in this way, tooth *d* being thus completely formed. It will be seen that obtaining accurate teeth by this method requires—first, an accurate templet; second, accurate setting of the templet and tool in proper relation to each other; and third, a bearing surface on the guide of exactly the same shape as the cutting edge of the tool. As shown, the gear to be cut has had the tooth spaces roughed out to shape, so that the finishing operation removes a comparatively small amount of metal.

The Odontographic Principle: In shaping teeth by the odontographic principle, the tool is guided in some way by suitable mechanism, to closely approximate the desired tooth outline by means of circular arcs, or other easily obtained curves. A simple example is shown in Fig. 3. The gear to be cut is held and indexed as in the two previous cases. The blank has had the teeth roughed out as in the previous case. The gear to be cut has involute teeth. With teeth of this form, in most cases a circular arc may be found which will more or less closely approximate the true outline. Such a circular arc is shown at $x\,y$, with its center at o. The radius tool holder shown has its center at o to agree with that of arc $x\,y$. The



Figs. 7, 8, 9 and 10. The Four Methods of Operation, as applied to the Molding-Generating Principle of Action.

cutting point of the tool used is located on arc $x\,y$. It will be seen from this, that when the radius tool is fed from position T_1 to T_2 by the feed worm, its point will follow the desired arc and cut the desired outline for the tooth. By this means, the upper surface of tooth *a* is formed. The same surfaces of teeth *b*, *c* and *d* have previously been cut, as well as the opposite faces of *d* and *e*, tooth *d* being completed. To cut the opposite faces, the work may be reversed on the arbor.

The Describing-Generating Principle: This principle is shown in Fig. 4, applied to the shaping of involute teeth. The cutting of involute teeth only has been hitherto shown in these examples, owing to the fact that in other cases, as in this, it lends itself most readily to the purposes of illustration. The involute, as is well known, is the curve formed by a point in a cord which is being unwrapped from the periphery of a circle. In the cut, the dotted line $x\,y$ shows an involute generated in this fashion from the *base circle* shown. This base circle is formed by the periphery of the rolling disk, which is firmly connected with the gear to be cut through the work arbor on which both are mounted. Unlike the previous cases considered, the work arbor in this case is free to revolve on centers without being restrained by an indexing mechanism; as in previous cases, the blank has had the teeth

roughed out. The tool used is a shaper, as before. To some fixed part of the machine is clamped the tape holder shown. This has fastened to it two thin flexible metallic tapes, M_1 and M_2 the former stretched between screw S_1 on the tape holder and the corresponding screw on the rolling disk, while the latter is similarly stretched between screws S_2 and S_2. By this means, it will be seen that when the shaper table is fed in the direction indicated, the unwinding of M_1 and the winding of M_2 will positively roll the disk and the work with it. If now, a tool be placed in the tool block of the shaper, having a cutting point set at the same height as the middle thickness of the steel tapes, and if the table be fed as shown, the mechanism will constrain the tool point to cut an involute on the side of the tooth of the gear blank. When the tooth is at *c*, the tool will be at T_c; when the tooth is at *b*, the tool, at T_b, will have cut down about half the length of the face, as shown; when the tooth is at *a*, its outline will have been completed on that side by the tool, at T_a. The way in which the involute is generated will be easily understood, when it is seen that the cutting point of the tool always coincides with a given point *y* in tape M_2, so that the same involute as is generated by this point in the unwinding tape is reproduced by the tool point. The device is incomplete, as shown, in that no provision is made for indexing. In this case the gear to be cut and the rolling disk have to be indexed with relation to each other, so as to present the different teeth properly for the tool to act upon them. At *d* is shown a completed tooth.

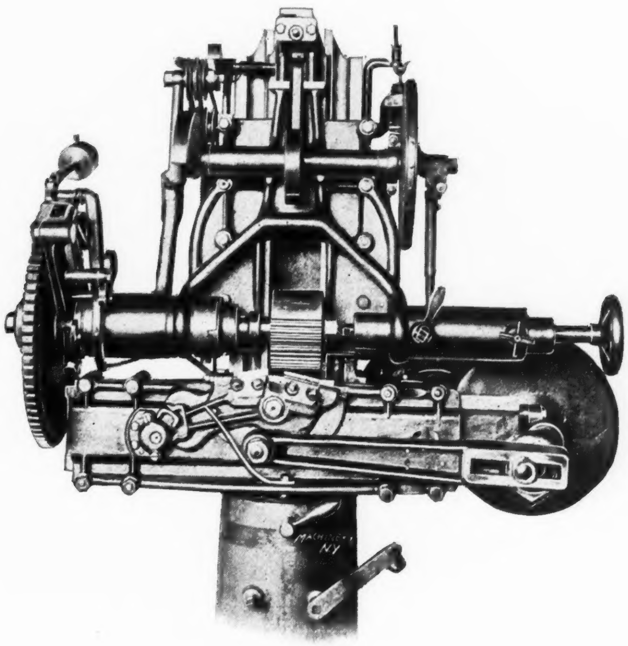


Fig. 11. The Pederson Formed Tool Gear Shaping Machine.

The Molding-Generating Principle: This method of making gears depends on the fact that in a set of interchangeable gearing a gear formed correctly to run with one of the series will run with all of the series. The molding process consists in using a completed gear tooth or gear, of proper shape, to from the others. Two examples of this are shown in Figs. 5 and 6. The first case supposes a forming gear, as shown, of correct shape. The blank to be formed is made of some plastic material like wax or clay. The blank and the forming gear are mounted on arbors at the proper distance apart, and rotated together at the proper speed ratio. The teeth of the forming gear, pressing into the plastic blank, will form spaces and press out teeth of the correct shape to mesh with itself, or with any other gear of the same interchangeable series.

In Fig. 6 the blank is of metal or other non-plastic material, and the forming gear is replaced with a forming cutter, having sharp edges of exactly the same outline. The blank, which in this case is of the full outside diameter of the gear into which it is to be made, is rotated with the cutter, as in Fig. 5. The cutter is reciprocated in the direction of its axis so as to take a series of cuts, to form the tooth spaces as the rotation takes place. The principle is identical with that shown in Fig. 5. Of course, the cutter has to be fed directly in to the proper depth to start with, before the rotating commences.

Four Methods of Operation.

In classifying gear-cutting methods by the operations involved, we will take for the purpose of illustration the molding-generating method as applied to the spur gear. Later on we will see how the same operations are applied to the cutting of other forms of gears, by other methods. In the four cases shown in Figs. 7 to 10, the molding-generating is done by a rack working in a gear, not by one gear working in another, as in Figs. 5 and 6.

By Impression: Fig. 5 is an example of this kind, the teeth in the plastic blank being formed by the impression made by them on the forming gear. In Fig. 7 the same thing is shown, except that the forming member is a rack which has shaped the periphery of the gear with which it meshes into correct teeth, as shown.

By Shaping or Planing: In Fig. 8 but one tooth space of the gear is formed at a time, and instead of using a rack to do the forming, a tool T_1 may be used having an outline the shape of a rack tooth. This is fed along horizontally, and the gear to be cut is rotated in unison with it, the same way as in Fig. 7. If tool T_1 is given a cutting movement in a shaper, the spaces formed will be of exactly the right shape and identical with those formed in the previous case. Each of the spaces will have to be formed in the same way one after another, the work being indexed with reference to the imaginary rack, to bring the tool into the proper position for each of them. Instead of forming both sides of a space at one operation, as with tool T_1, a single side tool T_2 may be used, corresponding with one side only of the rack. In this case one side only of each tooth is finished, so the tool or the work has to be reversed, after which the other sides are completed.

By Milling: Instead of using a planer or shaper tool to match the side of the imaginary rack tooth, a milling cutter may be used, as shown in Fig. 9. In this case the gear is rotated, and the milling cutter advanced to agree with the advance of the imaginary rack. The cutting face of the cutter must of course be formed on a plane surface, as shown. This arrangement presents some difficulties when the gear to be cut has a wide face, since the circular cutter will cut deeper into the tooth space at the center than it will toward the edges. This deepening of the tooth space at the center does not affect the acting tooth surface, and so is harmless (except possibly in the case of the generation of pinions having a small number of teeth, and involute outlines of low pressure angle, in which case the trouble due to interference is aggravated). The larger the diameter of the cutter as compared with the face of the gear, the less is the trouble on this score.

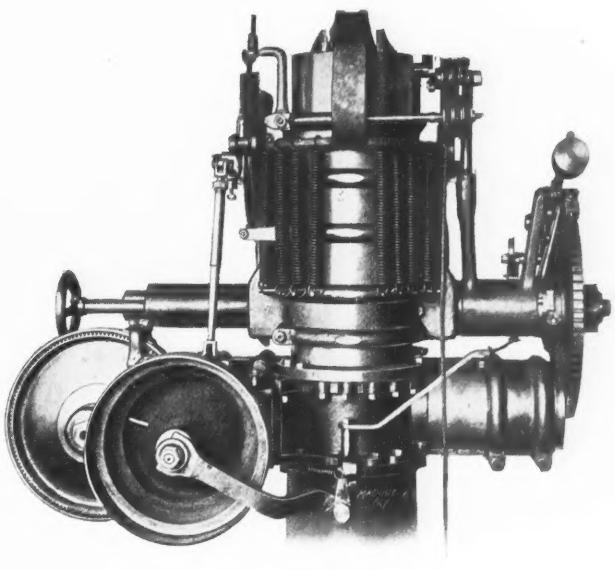


Fig. 12. Rear View of the Pederson Machine.

By Grinding or Abrasion: In Fig. 10, the milling cutter of Fig. 9 has been replaced by an emery wheel of similar shape, having a plane face perpendicular to the axis of the wheel

spindle. The action on the work is identical with that in the previous case, subject only to the limitations of the grinding process, such as the rapid wearing away of the material of the wheel, involving the necessity for constantly truing it up. Besides this, only a small amount of stock can be removed in a given time, as compared with the execution possible with a milling cutter. The process has the advantage that it can be used in hardened work.

As intimated, each of these various operations can be applied to different kinds of gears, acting according to different principles, though many of the possible combinations are impracticable. This preliminary discussion of methods, however, will serve to systematize the study of the various machines illustrated and described in the following pages, and will render an understanding of their construction more easily intelligible.

MACHINES FOR FORMING THE TEETH OF SPUR GEARS.

As described in the previous section, spur gear teeth may be formed in any one of five ways—by the formed tool method, the templet method, the odontographic method, the describing-generating method, or the molding-generating method. The extent to which these various schemes have been applied in practical use varies greatly. The formed tool method is at once the most obvious and the most used of them all. The templet principle has been applied to a limited extent, principally for gears of very large size. So far as the writer is aware, no practical application of the odontographic principle has been made in the cutting of spur gears. The only machine that has come to his notice involving the describing-generating process, was one invented by Mr. Ambrose Swasey, and in use a number of years ago in the shops of the Pratt & Whitney Co. This was not used, however, for making gear teeth, but for making gear tooth cutters—before the days of the formed cutter, which it was not adapted to making. The molding-generating process in various forms has received a wide application, second only to the formed tool method.

The operations available for the formed tool method are—impression, shaping or planing, milling, and grinding or abrasion. Of these, the impression process is obviously unsuited for practical work. The shaping or planing, and the milling (particularly the latter) have a wide range of application.

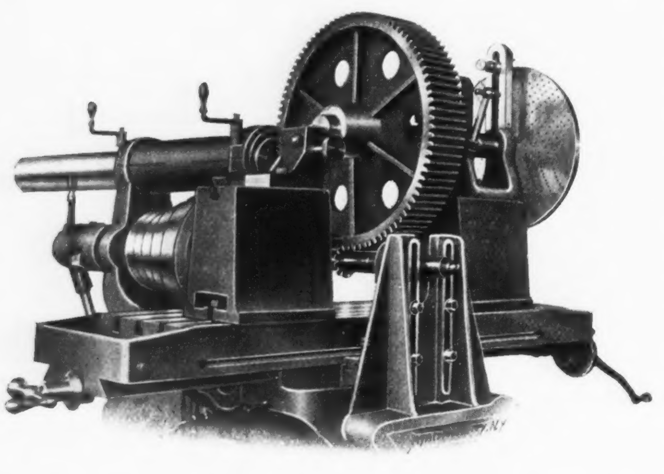


Fig. 13. Attachment for cutting Gears of Large Diameter on Cincinnati Milling Machines.

In the case of the operation of grinding or abrasion, but a single machine has ever been built embodying the formed tool principle, so far as the writer is aware.

Machines using Formed Shaper or Planer Tools.

The primitive application of the formed tool method is that in which a gear blank is mounted on index centers on the planer or shaper table, and has its teeth cut by a tool having an outline corresponding to the desired tooth space. In this operation the tool is fed by hand to the proper depth and withdrawn. The work is then indexed for a second cut, the tool is fed down again, and the operation is repeated until the gear is finished. This is shown diagrammatically in the upper part of Fig. 1. It is the simplest method of cutting a gear which has to be made immediately, and for which formed milling cutters are not available. It also has its application in the case of gears of unusual size. Under these circumstances, however, the machine used is generally a slotter instead of a planer or shaper. A formed tool is fastened in the tool-post of the machine, while the work is clamped to the revolving table. The indexing is done by such means as may be provided, usually a worm and worm gear or a master wheel.* The Gleason and Newton templet machines (described in the next installment) also may be, and doubtless often are, used in the same way.

Figs. 11 and 12 show a machine using the formed tool with the shaper method of action. The machine is an interesting one in its details, and it would require considerable space to

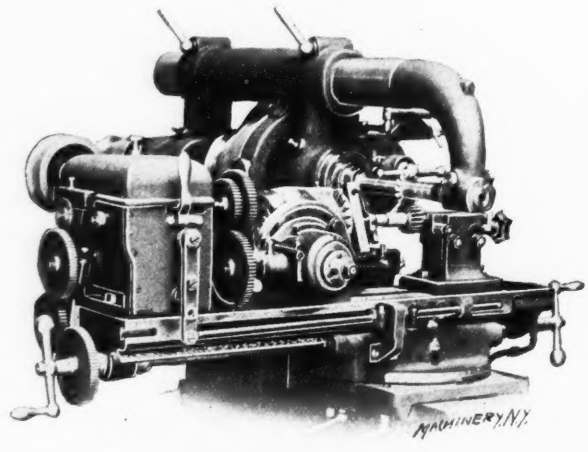


Fig. 14. Attachment made by Ludwig, Loewe & Co. for converting the Milling Machine into an Automatic Gear Cutter.

go into full particulars, so only a general description of it will be given. The mechanism is mounted on a circular column. The work arbor is carried by a slide, vertically adjustable to suit the diameter of the work, the adjustment being obtained by the crank handle shown at the front of the column. The feed and indexing movements for the work are controlled by cams on the shaft shown at the upper side of the work slide, this cam shaft being operated by an adjustable ratchet mechanism from the driving gear. One of these cams, by suitable lever connections, feeds the blank slowly down toward the cutters until the proper depth has been reached, when it allows the springs shown at the rear of the column to quickly return it, whereupon the cam at the left of the shaft operates the mechanism by which the work is indexed. The feeding cam then comes into action again—and so on until the work is completed. The depth of the feed given the work slide by the cam movement is varied by altering the position of the contact block by which the cam lever transmits the feeding movement to the slide. This is changed by the horizontal square-head screw seen at the extreme top of the column. The indexing is effected by a notched dial plate, operated by pawls and locking levers moved by the cam previously mentioned. The number of notches moved at each indexing can be regulated by a guard covering more or less of the teeth included in the stroke of the indexing pawl. Provision is made for stopping the action of the machine automatically when the required number of teeth have been cut.

The cutter slide is driven by a back-geared crank movement, adjustable for length of stroke and for various numbers of strokes per minute. Two tools are used, one cutting on the forward, the other on the return stroke. These tools, as may be seen from Fig. 11, are mounted in a rocking holder, which is tipped to bring first one and then the other into action as the end of each stroke is reached. This tipping is effected by the rocking and locking cam at the left end of the cutter slide in Fig. 11. This rocking and locking cam is connected with a slot cam near the right-hand end of the slide, this latter being operated by a pin near the crank end of the connecting-rod. As the connecting-rod passes the center, going in either direction, it operates the slot cam, which, through its connection with the rocking and locking cam, brings the desired

* See "Generating a Large Index Plate," by A. L. DeLeeuw, August, 1905.

one of the two blades into action. The one of these blades which has the heaviest of the cutting to do is of a simple U-shape, forming the bottoms and fillets of the tooth spaces. The other one, which has a lighter cut, forms the curved faces of the teeth.

Among the advantages claimed for this machine are extreme rigidity of action, exceeding that of the ordinary automatic gear cutter, and very low first cost. The cost of the cutters is also very moderate, being about one-fifth of that for formed milling cutters of the same pitch. These cutter blades are planed to shape, and may be ground on the face without change of contour. By means of special cutters straddling the teeth of the gear, provision is made for cutting pinions of few teeth and considerable under-cut. We are informed that the British rights have been acquired by Vickers Sons and Maxim, who are preparing to manufacture it on a large scale.*

Standard Machine Tools and Attachments Using Formed Milling Cutters.

More gears are cut by formed milling cutters than in any other way. It is distinctly a commercially successful process. The cutting tools are comparatively inexpensive, and retain

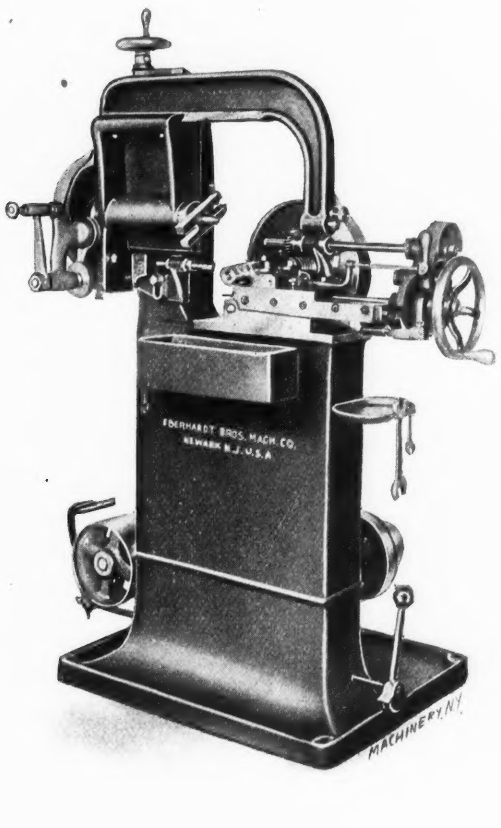


Fig 15. Eberhardt Bros. Semi-automatic Gear Cutter for Small Work.

their shape until they are entirely ground away, which is only after the accomplishment of a surprising amount of work.

The simplest way to use a formed cutter is in the milling machine. In a milling machine provided with an indexing head, no attachments are required for gears of moderate size and small pitch, and many thousands of them are cut with this simple equipment. For gears of larger diameter, though still of a pitch small enough so as to be within the range of the pulling power of the spindle, the worm-wheel of the dividing head becomes too small to satisfactorily and accurately index the wheel. For such work, many of the milling machine makers provide indexing attachments suitable for work of greater diameter than is possible otherwise. In Fig. 13 is shown an outfit of this kind built by the Cincinnati Milling Machine Co., Cincinnati, Ohio. The head- and foot-stock are mounted on elevating blocks which make it possible for them to swing work of a larger diameter. When working on large diameters, the table is raised and the cut taken on the under side of the work. This brings the thrust due to the cut down nearer to the bearing surfaces which have to resist it, and gives a steadier cutting action than would be the case if the work were lowered far enough to have the cutter act on the top of the blank. The indexing is done simply and directly by a plate with rows of holes, of numbers corresponding to the number of teeth it is desired to cut. This plate is of much greater diameter than the index worm of the regular spiral head, and so gives more accurate results.

An attachment of a different kind for cutting gears in the milling machine is shown in Fig. 14. Here we have an arrangement which is bolted on the milling machine table and connected with the dividing head. This attachment is driven from the counter-shaft by a special belt connection which serves to operate the feed and indexing of the work, the usual feed connections being disconnected. The device renders the milling machine automatic in all its actions. The table with the work on it is fed forward slowly until the cutter has passed through the work and formed the tooth space. The table and work are then rapidly returned, when the work is indexed and again fed forward as before. These processes are repeated until the gear is finished. The milling machine is thus made in effect an automatic gear cutter, capable of cutting bevel gears and clutches, as well as spur gears. This device is made by Ludwig, Loewe & Co., Berlin, Germany.*

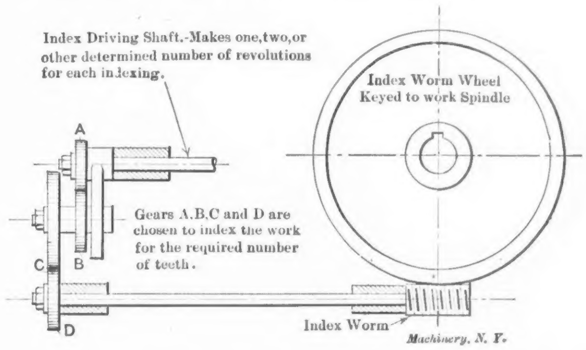


Fig. 16. Diagram showing the arrangement of the Standard Indexing Mechanism.

Semi-automatic Machines Using Formed Milling Cutters.

Leaving the special use of the standard milling machine in this work, and coming to milling machines specially adapted to cutting gear teeth, we are met by a bewildering variety of designs of varying degrees of ingenuity and interest. We will first consider the simpler forms of these specialized milling machines, or "gear-cutting machines," as we may better call them.

In the simpler forms, the development from the milling machine consists principally in embodying the dividing mechanism as a part of the machine, instead of making it an attachment. The feed may be operated by hand, or it may be connected by belt or gearing with the spindle, so as to be driven positively. In the latter case, an automatic stop is provided for throwing the feed out when the cut is completed. In the automatic form of machine the indexing mechanism, as well, is operated by power, as is also the quick return of the feed; and the movements are made dependent on each other

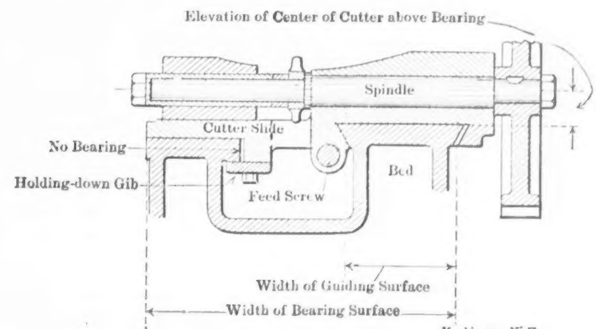


Fig. 17. Cross-section through Cutter Slide of Typical Gear Cutter, illustrating Certain Principles relating to Accuracy and Cutting Power.

* Information concerning the American rights for this machine, may be obtained from Mr. H. A. Elliott, with Alfred H. Schutte, corner of Cedar and West Streets, New York.

* See article entitled "Automatic Indexing and Feed Attachment for the Milling Machine" in the June, 1907, issue of MACHINERY.

in such a way that the machine of itself feeds the cutter through the work, returns it when the cut has been completed, indexes the work, and repeats the cycle until the job is finished.

An example of the semi-automatic form of the machine, made by Eberhardt Bros. Machine Co., Newark, N. J., is shown in Fig. 15. The mechanism is quite simple and may be readily understood from the cut. The spindle is carried on a slide which may be adjusted at an angle for cutting bevel gears, though it is shown in the cut down in a horizontal position as required for cutting spur gears. An automatic feed is provided for the spindle slide, connected with the cutter spindle by spiral gears and change gears, which may be set to give the desired rate. An automatic stop is provided for throwing out the power feed when the required length of cut has been taken. The slide then has to be run back by hand and the work indexed by hand, when the automatic feed is again thrown out. The work spindle is carried on a slide gibbed to the face of the column of the machine. This slide carries the indexing mechanism also, and an over-hanging arm for supporting the outer end of the work arbor. The indexing mechanism is of the same type as that illustrated in Fig. 16 and described later. Such tools are adapted to manufacturing in small quantities where unskilled labor is employed. The machines are inexpensive, and the operating skill required is of a comparatively low order.

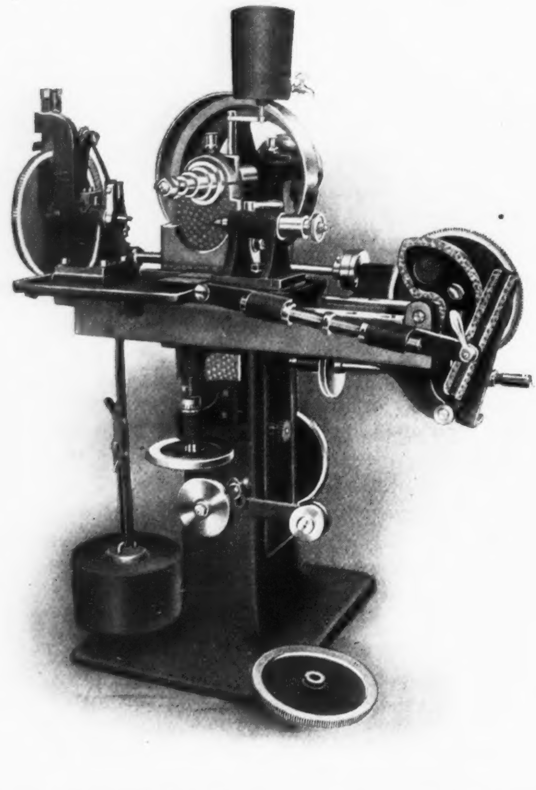


Fig. 18. The Dwight Slate Cam Actuated Automatic Gear Cutter.

Machines of similarly simple action, but for larger work, have been built from time to time by builders of special machinery as required by their customers.* These machines may be handled by comparatively inexpensive labor, are furnished at a low first cost, and are very substantial in construction.

Automatic Machines using Formed Milling Cutters—General Principles of Design.

The fact that we illustrate twenty-five automatic cutting gear machines of this type, built by twenty-two makers, is good evidence of its commercial position. Much thought and experience has gone into the development of the automatic gear cutter. In selecting a machine of this kind, important requirements to be looked out for are—accuracy of indexing, power and durability of the feed and cutter-driving mechanisms, rigidity of construction, convenience of handling, and range of usefulness.

* See, for instance, the description of Newton machine, in article, "American Gear Cutting Machinery," in the June, 1898, issue of MACHINERY.

In the matter of accurate indexing (which is of prime importance, especially for gears which are to run at high speeds), the important considerations are the accuracy of the index worm-wheel and the mechanical construction of the indexing mechanism. With the exception of the machines shown in Figs. 18 and 19 (also 41, 42, 43 and 44, shown in the next installment), which are for comparatively small work with small numbers of teeth, the principle of the indexing mechanism is the same for all of these machines.

The work spindle has mounted on it (see Fig. 16) a worm-wheel driven by an indexing worm. This worm is connected by change gears *A*, *B*, *C* and *D* with a shaft which is arranged (usually) to make one complete revolution, when the proper time for indexing arrives. The change gears are so set in connection with the invariable movement of the index shaft, as to give the exact movement required to rotate the blank to the point where it is desired to cut the next tooth. In some machines, provision is made for giving two or four complete revolutions to the driving shafts when the number of teeth to be indexed is small. It is important that the mechanism by which this shaft is set in motion and stopped shall be very carefully designed, so that the stopping will always take place at exactly the same point in the rotation, thus permitting no over-running or under-running of the worm.

In the construction of the worm-wheel, there are two plans followed. Some makers, notably the Brown & Sharpe Mfg. Co., prefer to make each worm-wheel an accurate copy of a master wheel which they know to be of unimpeachable accuracy. Other builders prefer to make each index wheel by itself, and generate each one to a high degree of perfection by methods well understood and commonly employed in such work, generally involving making the rim in halves.

In the matter of obtaining power and durability for the drive, a variety of opinions will be found expressed in the various designs. Some of them have the spindles driven by spur gearing (or bevel gearing in some cases) while other makers prefer spiral or worm gear drives. There is much conflict of opinion as to the advantages of these various forms. In some cases the builder is restricted in his choice by structural features which limit him to one form only. In any event, the drive should be smooth and powerful.

The capacity of gear-cutting machines for taking heavy chips may be easily allowed to fall below the limit of the driving power available at the spindle, if the frame of the machine and the design of gibbing of the various slides are such as to make the machine lacking in rigidity. The various requirements for doing work rapidly and accurately may be understood from the rough sketch of a section through the spindle and cutter slide of a gear cutting machine shown in Fig. 17. This sketch does not represent any particular machine, but shows some features which are common to a number of makes; unnecessary details have been omitted. One of the requirements is that the strain of the cutting action shall be brought as close to the bearing surface as possible. This applies in the case of both the work spindle and the cutter spindle. It will be noted in Fig. 17 that the elevation of the center of the cutter above the bearing is very small,

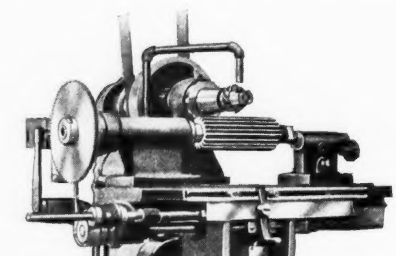


Fig. 19. The Sloan & Chace Automatic Gear Cutter for Small and Medium Work.

so that the irregular thrust of the cutting action when working at full capacity has little effect in disturbing the rigidity of the machine. Bearing surfaces of great area are also advisable to give firmness to the structure. It will be noted that the bearing surface extends the full width of the bed of the machine in the sketch. With this wide bearing surface, however, provision should be made for guiding the slide for alignment, with much narrower surfaces, to prevent a cramping or "bureau-drawer" action, as it has been called. The guiding in this case is done entirely by the comparatively narrow dove-tail slide at the left. There is no side bearing at the

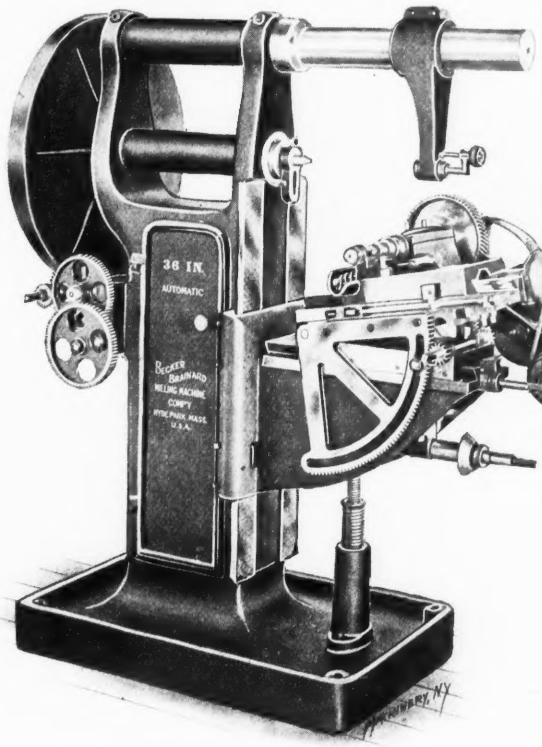


Fig. 20. The Becker-Brainard Automatic Gear Cutting Machine.

right, a clearance space being left at the right hand edge of that slide as indicated. A strap is provided here, however, for holding the slide down onto its bearing, thus preventing any lifting tendency which may develop at this point. It will be noted that the feed screw is placed quite close to the cutting point. This provision also tends toward smoothness and ease of action, since the power is then directly applied instead of being applied in a way to cramp the slide on its bearings. The rigidity and smoothness which these provisions insure are of great importance in permitting the use of very heavy cuts and lengthening the life of the cutter.

In regard to these matters, and the matter of convenience of operation as well, much can be surmised by the mechanic from a careful inspection of the engravings we show herewith. Instead of making invidious comparisons in these particulars, it has been thought best to let the reader draw such conclusions as he can from the information given. The descriptions of the various automatic gear-cutters will be found to contain explanations of their construction, and to refer to such particular points as may be peculiar to each case. To make comparisons easy, machines of similar type have been placed together in regular order. Of course such of the good qualities in a machine tool as depend on accurate workmanship and the design of details not visible from the exterior, will have to be judged by other means than a mere inspection of half-tones. In the matter of accurate workmanship, particularly, the reputation of the builder will go a long way with the intending purchaser.

Miscellaneous Forms of the Automatic Spur Gear Cutter.

As has been stated, the automatic gear cutter is a specialized form of the milling machine. There are no machines in our list that show this more plainly than the two shown in Figs. 18 and 19. The first of these is built by the Dwight Slate Machine Co., of Hartford, Conn. The machine at once shows itself to be a modified milling machine, with the usual screw feed replaced by a cam mechanism which gives a slow forward movement and a quick return. This is altered to give the proper length of feed, by means of the slotted link shown. The orthodox dividing head with worm and worm-wheel, has been replaced by a dividing plate on the head-stock spindle, with notches to correspond with the number of teeth it is desired to cut. An automatic trip is provided which throws out the feed at the completion of the last tooth. The various adjustments for different diameters of gears and lengths of cut will be readily understood from an inspection of the figure. This machine is one of a series of three of varying sizes, one of which is adapted to the cutting of bevel gears as well as spur gears.

In the machine shown in Fig. 19, built by Sloan & Chace Manufacturing Co., Ltd., Newark, N. J., the feed is effected by a screw, as in the ordinary milling machine, instead of by cams as in the previous case. The motion for the indexing and quick return is taken from the counter-shaft by the pulley shown near the base of the machine at the left. This gives a constant speed at the highest practicable rate whatever the spindle speed may be. The cutting feed is obtained from a connection with the spindle through a feed box, giving three changes. The indexing is ingeniously effected by the first half-turn of the feed screw, and is done positively without requiring the use of springs. A dial plate is used as in the previous case. The spindle head is adjustable in and out on the top of the column for centering the cutter. This machine shows the influence of the watch machinery makers' ideas, applied to a machine of rather larger capacity than usual with such construction. It is intended to show the watch makers' ideas of accuracy as well.

Another machine that shows the hereditary influence of the miller is shown in Fig. 20. In this case, however, the relative positions of the work spindle and the cutter spindle have been reversed from that occupied in the milling machine, or in the tools shown in the two preceding cuts. The work spindle passes through the uprights of the column, and car-

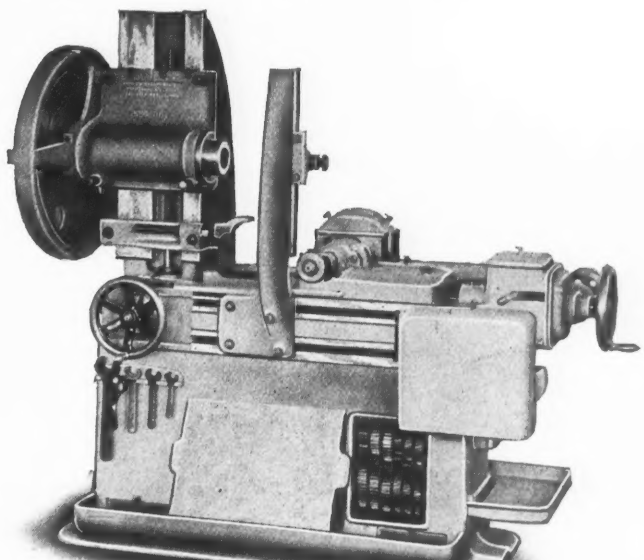


Fig. 21. An Example from the Brown & Sharpe Line of Automatic Gear Cutters.

ries a worm dividing gear at the rear end, while the cutter slide is located on the knee. In the position shown, with the intermediate quadrant elevated, the cut is being taken on an angle as would be required in cutting bevel gears, for which this machine is also adapted. For cutting spur gears, the slide is horizontal. Change gears for dividing are seen at the rear of the column beneath the casing for the indexing wheel. The indexing is done by a friction mechanism which is released at the proper time, coming up against a positive stop when one revolution has been made. The spindle driving

* Described in "New Machinery and Tools" section of MACHINERY, January, 1907.

and feed mechanisms are carried entirely by the knee. It will be noted that the gear driving the cutter spindle has helical teeth. Where a cutter spindle is to be driven by spur gears, this is a construction often followed, particularly in Europe, to give a smoother and more even motion than would be obtained by teeth cut straight across in the ordinary fashion. An incidental convenience of this machine is a trough just beneath the cutter spindle, enclosing a slowly-moving spiral conveyer. The chips fall from the cutter into the trough, and are pushed by the conveyer out over the edge of the knee into the pan base, away from the mechanism of the machine. This machine is built by the Becker-Brainard Milling Machine Co., Hyde Park, Mass.

The Orthodox Automatic Spur Gear Cutter.

The automatic gear cutters of the conventional type, for small and medium-sized work, have the work and cutter spindles both horizontal, and arranged in the same relation to each other as in the Becker-Brainard machine. Instead, however, of adjusting the machine for the diameter of work by the raising or lowering of the knee carrying the cutter spindle, the work arbor is raised or lowered, being carried for that purpose in a head vertically adjustable on a column at the end of the bed of the machine. The cutter slide on these machines, when arranged for cutting spur gears only, is gibbed directly to the top surface of the bed.

One of the best known examples of this orthodox type of automatic gear-cutting machine is that built by the Brown & Sharpe Mfg. Co., Providence, R. I. A front view of one of their smaller sizes, the No. 3, is shown in Fig. 21. The spindle of this machine is driven by worm gearing, though with the parts reversed from the order they would naturally take, since the worm-wheel is the driver, and the worm, which is much larger in diameter than the wheel, is the driven member.* This arrangement gives the smoothness of drive of worm gearing, an enlarged bearing area, and the advantage of being able to shift the whole spindle with its driving gear endwise in adjusting the cutter centrally with the work, instead of requiring that the driving gear remain fixed in position, driving the spindle by sliding keys, as in the ordinary

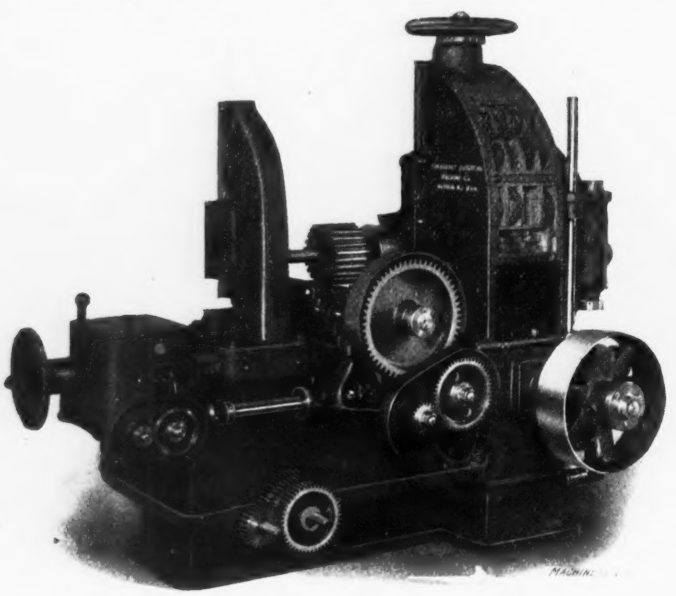


Fig 22. A Machine made by Eberhardt Bros., with Special Shortened Column; of Rugged Design for cutting Coarse Pitch Pinions.

construction. The plan generally followed by this company with all its machinery, of building the various parts of the mechanism on the unit system and assembling them as units in the machines, gives an air of neatness in design to the tool, which will be readily appreciated from the cut. A feature common with most automatic gear cutters, the outboard support for the work arbor, will be noticed clamped to guiding surfaces on the front of the bed. The index wheel is solid, and is made an accurate copy of a precision master wheel, as previously explained.

* Described in article entitled "The Figuring of Gear Drawings," by L. D. Burlingame, in engineering edition of MACHINERY, August, 1906.

The spur gear cutter shown in Fig. 22 is built by Eberhardt Bros. Machine Co., of Newark, N. J. The machine shown was designed primarily for cutting pinions of large pitch. For this reason the machine is ruggedly built and has a comparatively short column, limiting the diameter range for which it is adapted. The whole mechanism is driven from a pulley running at constant speed, the various changes of spindle speed and feed being obtained by change gears. It is like most other machines of its class, also, in the fact that the interior of the base of the machine serves as a collecting chamber for chips and a reservoir for the oil which is drawn from them.

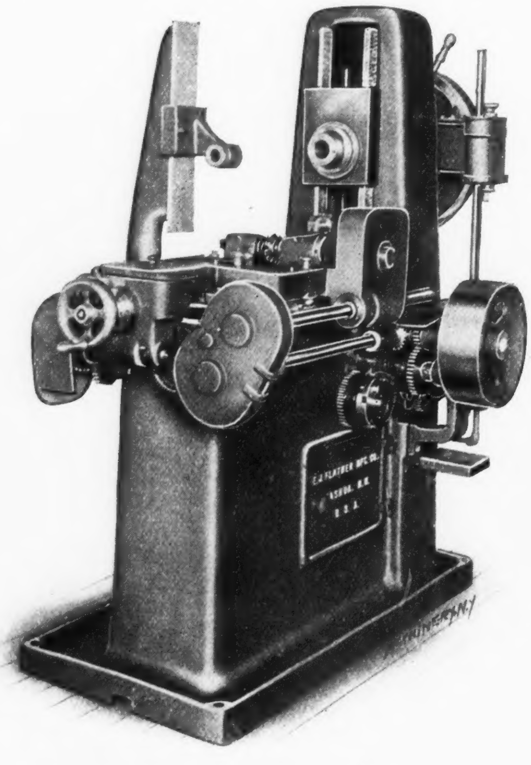


Fig. 23. The Flather Automatic Gear Cutter.

From here it is returned to the cutter by an oil pump. It will be seen that the cutter spindle driving gear in this case is a spur gear. The change gearing for altering the speed is placed next to it in the order of transmission, so that the splined shaft which leads the motion to the cutter shaft runs constantly at high velocity, whatever the speed of the cutter may be. Smaller machines of somewhat similar type are also built by this firm, some of them adjustable for cutting bevel gears as well as spur gears. The machine shown in Fig. 22 is, of course, ordinarily supplied with a higher column, than is here shown, which provides for work of greater diameter.*

An automatic gear cutter built by the E. J. Flather Mfg. Co. is shown in Fig. 23.† One of the most noticeable features of this machine as compared with those previously considered is the construction of the column which supports the work carrying head. This is made double, and the work carrying head passes through it instead of being clamped to ways on its face. The handle shown projecting at an angle in front of the index wheel casing at the back of the column, is used for clamping the work head solidly to its seat on both the front and back sides of the column, when the adjustment for depth of cut has been made. The spindle of this machine is worm-driven. The indexing mechanism is of the positively operated type, with a friction device to prevent rebound. As in previous cases, all changes of feed and speed are made by change gears, the machine being driven by a constant speed pulley.

* * *

A French formula for fireproofing wood and cloth calls for a mixture of $4\frac{1}{2}$ ounces of sulphate of ammonia, one-half ounce of borate of soda, one-sixth ounce of boric acid, all diluted in $2\frac{1}{4}$ pounds of water.

* See "New Machinery and Tools" section of MACHINERY, November, 1906, and June, 1907.

† See "New Tools of the Month" section of MACHINERY, July, 1904.

GIGANTIC METAL-WORKING PLANER.

The accompanying illustrations show the largest and heaviest metal-working planer ever built in America, and as far as we know, it exceeds also in size any tool of this class of European production. This planer was recently shipped from the Bement-Miles Works, Philadelphia, of the Niles-Bement-Pond Co., to the Mackintosh Hemphill Co., of Pittsburg. The best idea of the size of this planer is gained from the photograph reproduced on this page, which shows the planer erected in the shop, just prior to shipment, with 89 men distributed over it. If it is preferred to arrive at an idea of the size of this planer from figures, it may be mentioned that the total weight of this gigantic machine is 845,000 pounds (422½ tons), and that five motors, with a total power aggregating 207½ horse-power, are required for its operation. An idea of the difficulties attending the shipment of a machine of this size, as well as of the dimensions of the machine itself, may also be gained from the single fact that a special flat car with a central opening in the deck was required for the uprights alone, these being so large that their dimensions exceeded the clearance limits of tunnels and bridges, if loaded directly on the car deck in the usual position. The total weight of the two uprights is about 110,000 pounds.

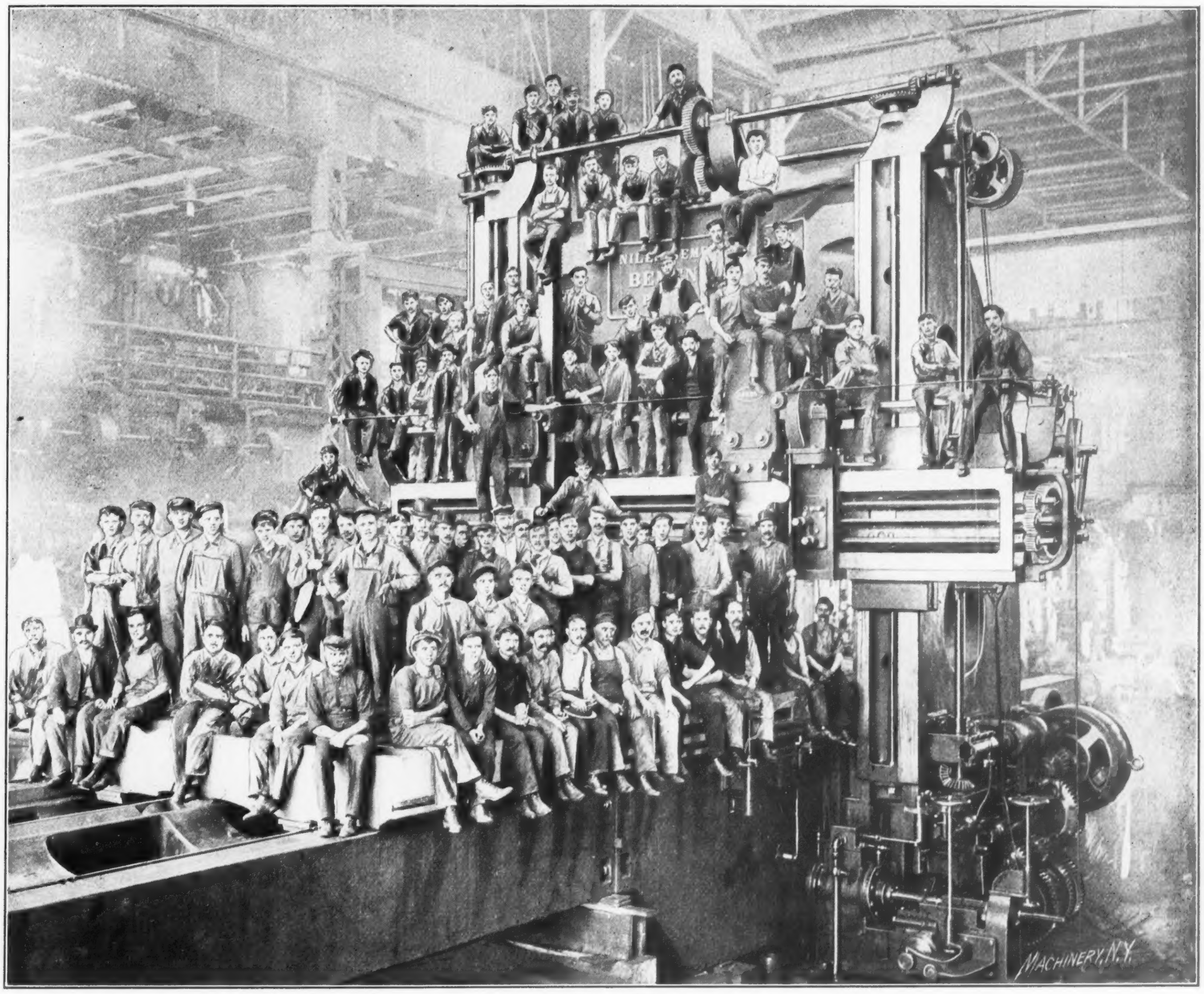


Fig. 1. Gigantic 14-foot Planer built at the Bement-Miles Works of the Niles-Bement-Pond Co.

General Dimensions.

The machine normally is a 14-foot planer of the ordinary type, as far as general principles of design is concerned, but in addition to the common features of planers, it is provided with some other features, which will be described later, and which are not found on general standard machines. The distance between the uprights is 14 feet 4 inches; the maximum distance from the table to the bottom of the cross-rail, 12 feet 2 inches; maximum stroke of table, 30 feet; total width of bed, 13 feet; length of bed, 60 feet. The table ways are each 15 inches wide. The tool slides are 7 feet 8 inches long, having a 4-foot vertical traverse. The cross-rail is long enough to admit full traverse of either head between the uprights. The face of each upright is 2 feet 6 inches in width, and the vertical height of the cross slide, including the top rib bracing, is 5 feet. Figs. 2 to 9 inclusive, which show the details of the planer, convey also some idea of the great size of the component parts.

Power Equipment.

As mentioned, the machine is provided with five motors. The main driving motor is of 100 horse-power, the lifting motor for the cross-rail is of 20 horse-power, and the traverse motor for the heads on the cross-rail, for fast traverse, is of 7½ horse-power. One motor of 50 horse-power normal capacity is provided for the independent slotter bars placed in each head, which are driven by rack and pinion. This motor also operates the heads for cross planing, the machine being adapted not only to ordinary planing, but to slotting and cross planing work as well, thus enabling the planer to be of service on the top, ends and sides of the work at one setting. The driving clutches are operated pneumatically, and an air compressor is provided, with a 30 horse-power motor, the planer being fitted with its own air compressor and independent motor. The planer is thus independent of the air supply of the shop, although the shop supply may be used, if it is desirable. Another feature of notable interest is a complete switchboard, which is furnished for controlling all the motors.

Provisions for Cross-planing and Slotting Work.

As has been mentioned, each head is fitted with a slotter bar independently driven by a rack, giving practically constant cutting speeds and a quick return. The maximum stroke

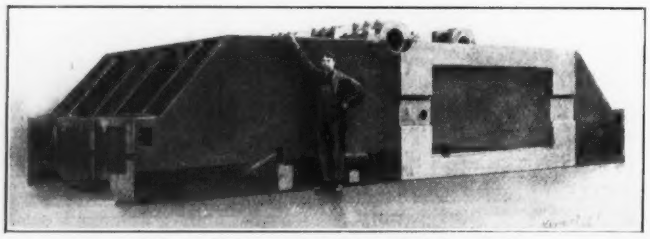


Fig. 2. Section of the Planer Bed, Inverted.

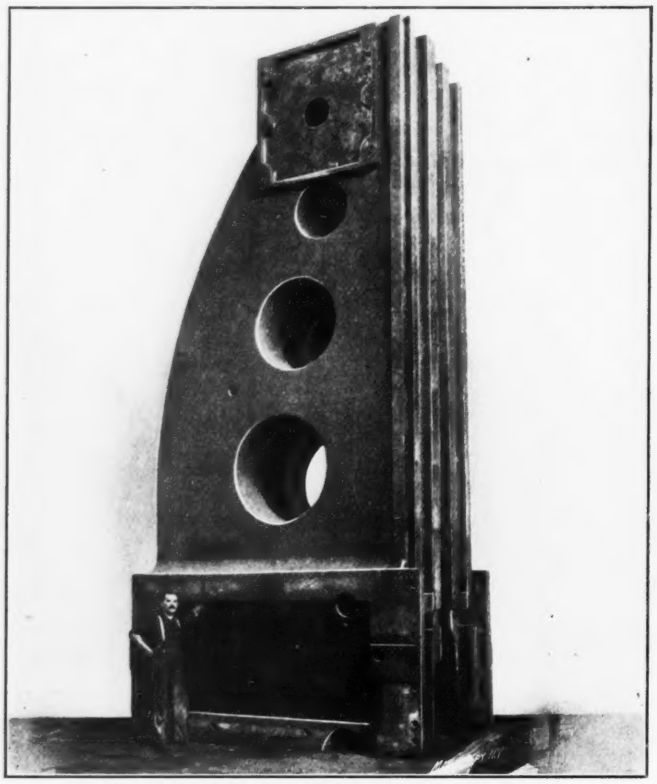


Fig. 4. The Upright before Assembling.

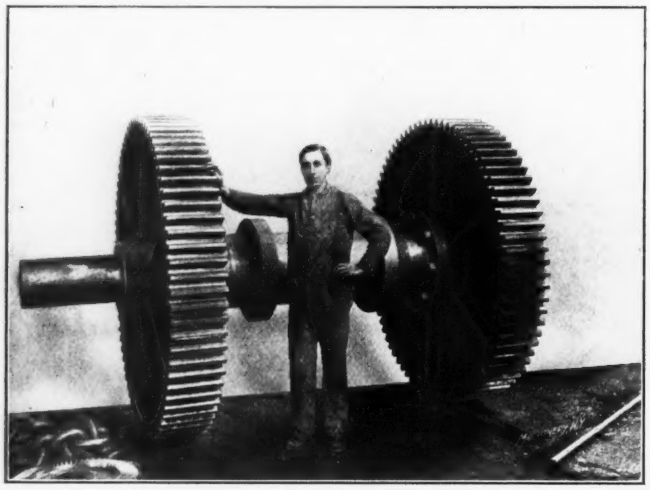


Fig. 6. Bull-wheels.

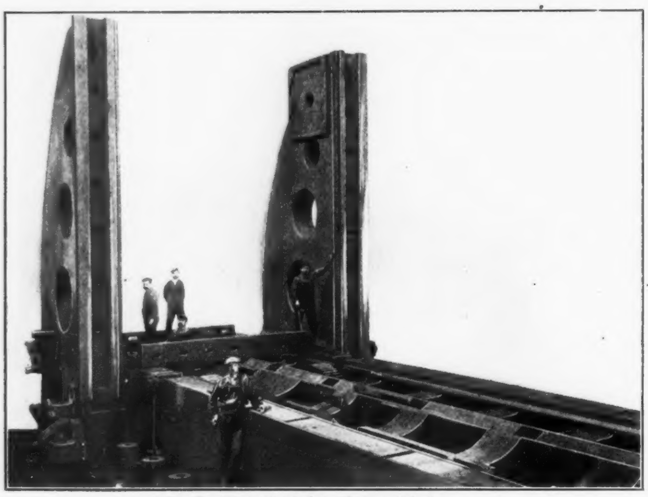


Fig. 8. The Uprights Assembled.


Fig. 3. Tie-beam, a Comparative View showing the Tie-beam's Unusual Proportions.

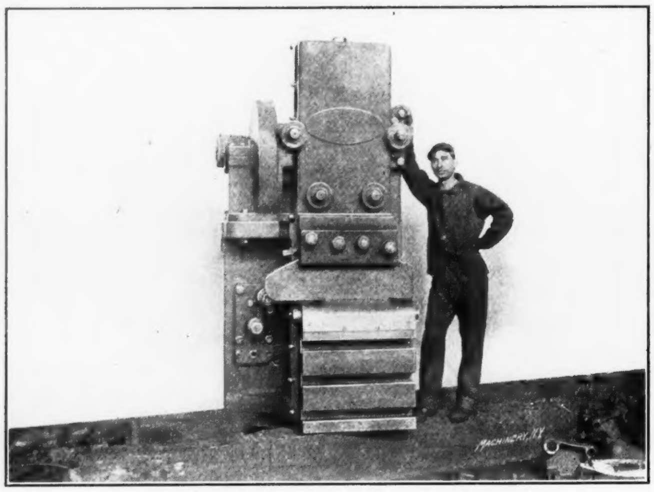


Fig. 5. View showing the Size of the Planer Head.

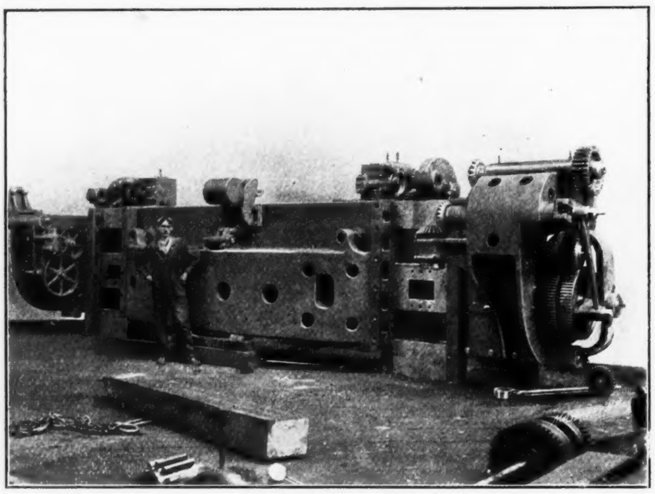


Fig. 7. Cross-rail.

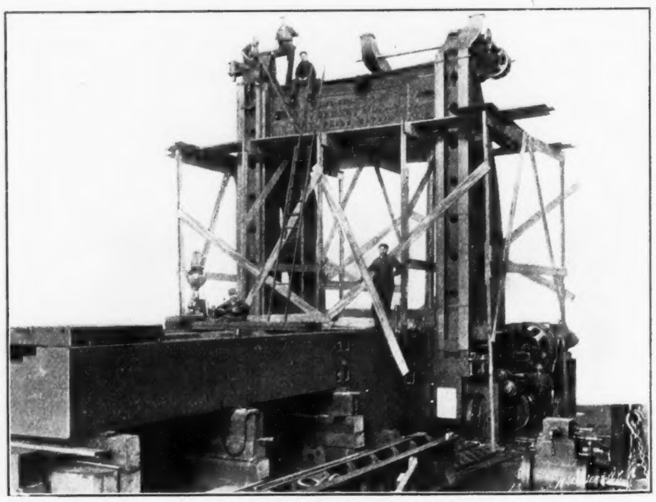


Fig. 9. Erecting the Tie-beam.

of the slotter bar is 8 feet. The cross planing movements of the heads, also referred to above, can be varied as to speed within certain limits, and are also provided with a quick return motion. The movements for slotting and transverse planing make it necessary to throw out the rack driving mechanism of the table. When the machine is used for slot-

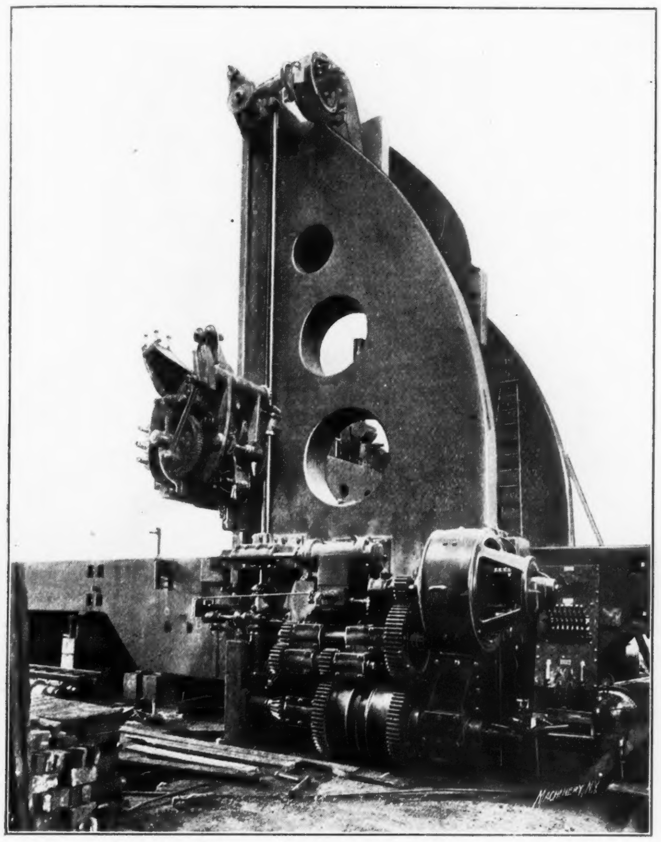


Fig. 10. View showing the 100-horse-power Motor for the Main Drive.

ting or transverse planing, the table is connected to a separate feed motion entirely distinct from the rack feed motion. The throwing out of the driving mechanism is accomplished by simply putting the pneumatic driving clutches into the idle position.

Speeds and Feeds.

The cutting and return speeds are variable through the motor, which has a variation of 1 to 1¼. Changes in speed may also be effected through a range of change gears. The regular cutting speeds are from 14 feet to 25 feet, and the

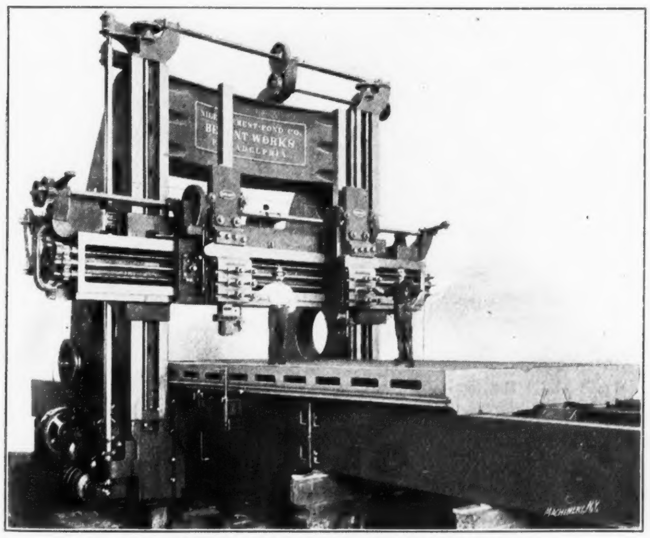


Fig. 11. View of the Slotter Drive.

return speeds are from 52½ to 65½ feet per minute. The slotter bars have a cutting speed of from 18½ to 30 feet, and a return speed of from 57 to 71 feet. Finally, the cutting speed for cross planing is from 11½ to 19 feet, and the return speeds from 35 feet to 43½ feet. The cross traverse feed of the heads is 50 inches per minute. The vertical speed for raising and lowering the cross rail is 26 inches per minute.

The Main Drive.

In Figs. 1 and 10 the main drive from the 100-horse-power motor is clearly shown. The drive is through gearing from the motor to the pneumatic reversing clutches at the base of the right-hand upright. These pneumatic clutches are completely encased, and are made, according to the Niles-Bement-

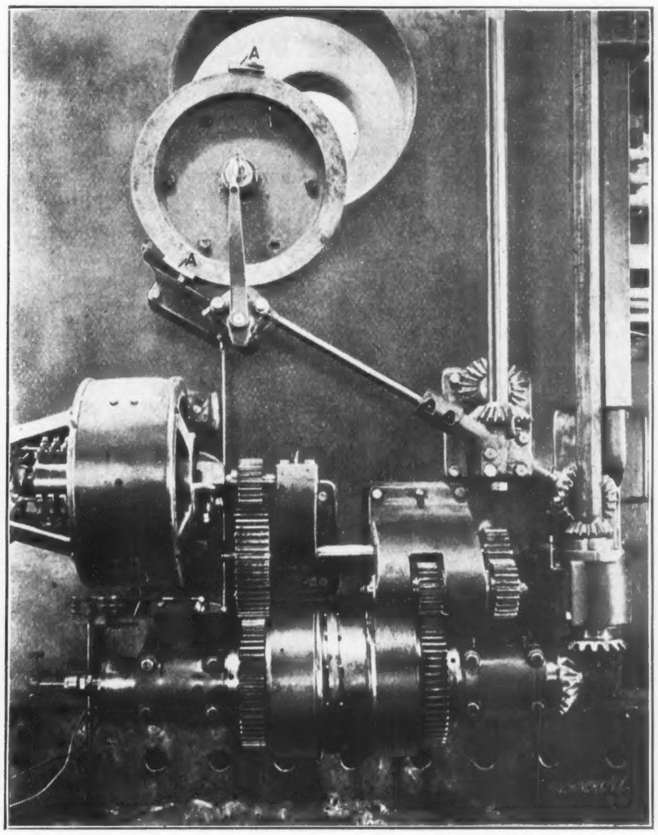


Fig 12. Detailed View of the Slotter Drive.

Pond type, with a large number of friction disks, through which arrangement a large friction area is obtained within a comparatively small space. The stopping, starting, and reversing of the table is controlled through a small valve easily moved by hand. The remaining portions of the drive, from this point on, are practically in every respect the same as those found on other planers, excepting, of course, that the drive distinguishes itself through its enormous proportions. All parts of the drive are made of steel, and the two bull pinions are forged directly on the shaft. They are cut so

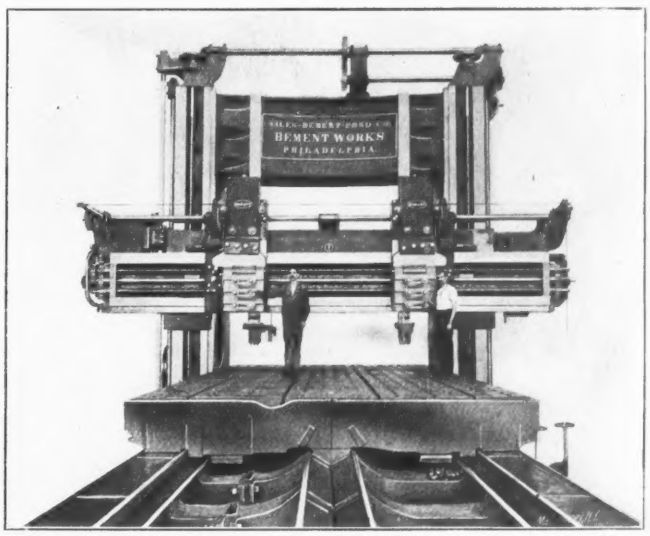


Fig. 13. Front View showing the Three Ways which support the Table.

that the teeth of the one are half a pitch ahead of the teeth in the other. This arrangement tends to produce smoothness of action.

The Pneumatic Feed.

The feed of the cross-heads is operated pneumatically. On the side of the upright, just above the gearing, Fig. 10, there is a cylinder with a piston rod extending to the left. In this

piston rod rack teeth are cut which mesh with a pinion near the lower end of the vertical feed shaft. Near the extreme end of this shaft is placed a bevel gear meshing with another bevel gear on a horizontal shaft, which latter transmits motion to the vertical feed shaft on the left-hand upright on the other side of the planer. The movement for these feed shafts is constant at all times, and the variations in amount and direction of the feeds of the heads are obtained by adjusting the connecting-rods in the slotted cranks on the ends of the cross-rail. In order that definite cross and vertical feeds may be obtained, these cranks are graduated. An angular feed can be obtained and given to the tool by using at the same time the slotted cranks for both vertical and cross feeds. This provision is also necessary on account of the fact that the heads are not designed to swivel. At each end of the stroke of the planer table the valve controlling the air for the feed cylinder is tripped automatically. The feed is thrown out by the closing of a valve on the main air supply pipe to the cylinder.

The table feed, which is used when slotting or planing transversely, was not put in place when the photograph from which Fig. 10 was reproduced, was taken, but it is shown in Fig. 14 directly in front of and at the base of the upright. The operation of this feed is practically the same as the operation of the feed for the cross-head, previously described, excepting that the amount of feed is obtained by varying the stroke of the piston in the cylinder, an adjustable stop being used for this purpose. This adjustment is made by the right-hand hand-wheel shown in Fig. 14, the left-hand hand-wheel serving the purpose of connecting and disconnecting this feed mechanism to and from the main driving shaft.

Slotter Drive.

In Fig. 11 is shown the slotter drive on the left-hand side of the planer, and in Fig. 12 this drive is shown in a larger scale. The general arrangement of this drive is the same as for the main drive on the opposite side, up to and including the pneumatic clutches. For the slotter drive the power is then transmitted through bevel gears to a vertical square shaft, from which the power in turn is transmitted to a horizontal square shaft running on the top of the cross-slide, as plainly shown in Fig. 11. A pinion on this shaft drives a large gear enclosed in the gear cover shown at the side of each head. On the same shaft as this large gear is a pinion which meshes with the rack teeth on the back of the slotter bar. The pinion on the square shaft slides back and forth on the shaft, and can be thrown out of mesh when desired, so that either one or both bars may be used. The disk or wheel shown directly above the motor on the left-hand upright controls the length of the stroke of the bar. This disk is driven from the vertical square shaft, and adjustable stops on the outside of its rim can be set at any desired point. These stops effect the reversal in the same way as the dogs on the side of the planer table effect the reversal of the table motion. Near the lower end of the vertical square shaft is seen a bevel gear on the end of a horizontal shaft through which motion is transmitted through the bed and to the other side of the planer, where the shaft is connected to the mechanism operating the valve of the feed cylinder on the opposite side, so that the feeds are available when using the slotter bar as well as when using the main drive for the planer.

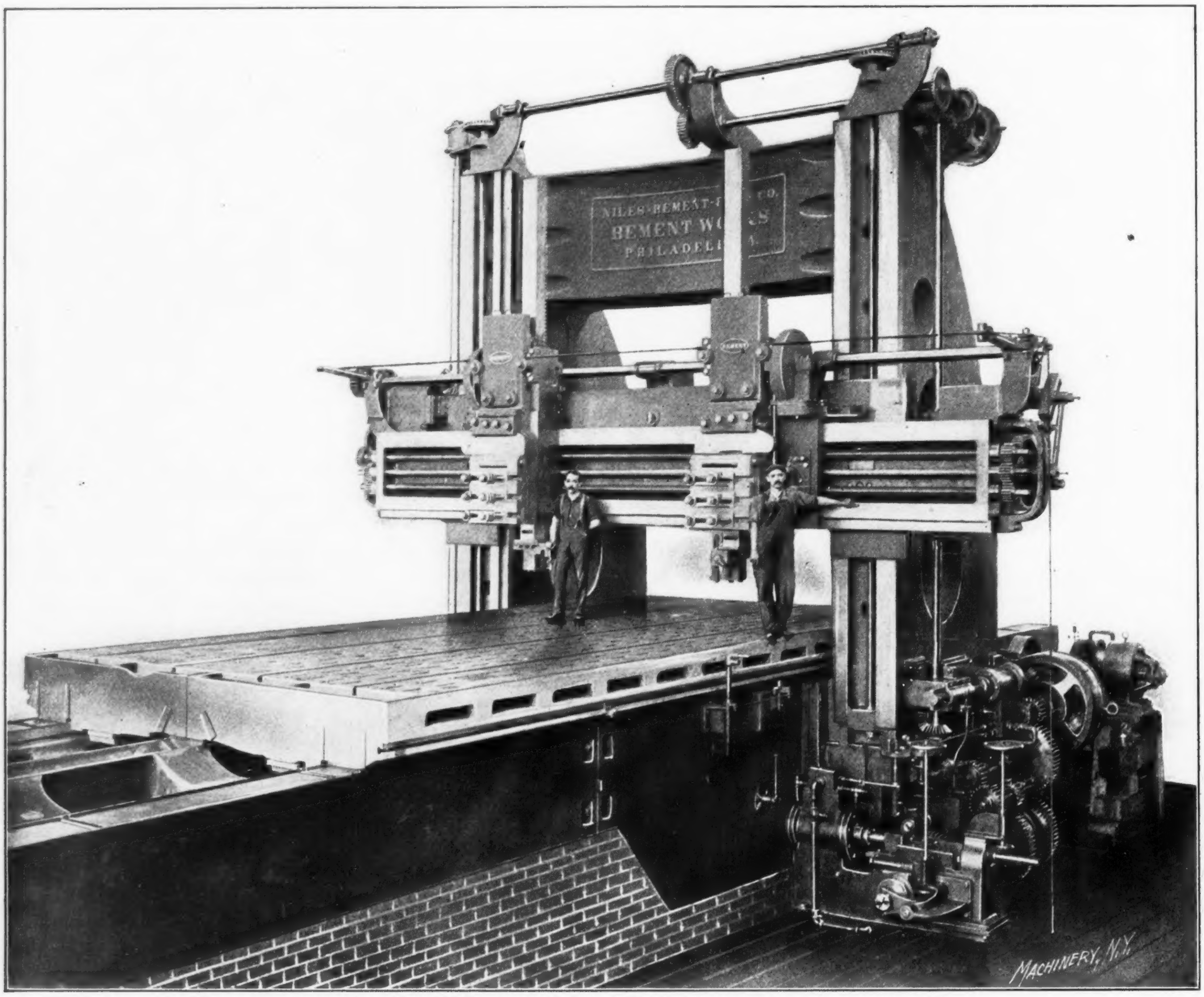


Fig. 14. General View of the Planer showing the Slotting Attachment and the Feed Mechanism.

Hand Control of Slotter.

In Fig. 14 a vertical shaft will be noticed (at the usual place for the reversing lever) the upper end of which is provided with two sockets, in one of which there is a handle. The upper of these sockets is connected to a shaft which runs down to the bottom lever or crank. This handle is for the hand control of the slotter. The method of connection can

be easily followed, starting from the handle shown in Fig. 14, going through the bed to the corresponding arrangement in Fig. 11, and then following the curved connecting-rod running in back of the upright and out through the upright to the slotter reversing mechanism at the reversing disk previously referred to.

The lower socket in this vertical shaft controls the movement of the table when the planer is used for ordinary planing. The mechanism is then connected by a lever and rods to the reversing dogs on both sides of the planer. It will be noticed that only one handle is furnished for each side of the machine. This prevents mistakes arising from throwing the wrong lever.

The Planer Bed.

It is evident that the bed for a planer of such dimensions as this could not very well be made in one piece, or, at least, an attempt to do so would have been impracticable. To produce and handle such a piece of work in the shop would not only have been attended with the greatest difficulties, if at all possible, but it would also have been a very difficult piece to ship. The bed is therefore made in seven parts, each end section being made in two parts, and the central section being made in three parts. The total weight of the bed is about 275,000 pounds.

The Table.

The table of the planer is made in two sections, the table being divided longitudinally in the center. The weight of the two sections together is about 140,000 pounds. The holes in the table for clamping bolts, etc., run entirely through an upper plate, but below this plate there is a second plate, without openings, extending the full width of the table. In the pocket formed between two plates of the table, all chips will collect. These can then easily be removed through the side openings of the table, clearly shown in Figs. 11 and 14. The table slides on the bed on two flat ways at the sides of the table, and in one V-way at the center, as is shown in Fig. 13.

At the end of the table, as shown in several of the cuts, but perhaps most plainly in Fig. 13, there are finished pads, two over the V-way and to two over each of the flat ways. These pads are intended to carry tool-heads when it is wanted to true up the ways, when worn out of alignment. The method of performing this work is rather interesting and, as far as we know, distinctly new. The table is raised about ¾ inch above the ways, and is supported in this position on sliding blocks which fit the narrow inner auxiliary ways, shown in Fig. 13, which are used for this purpose only. The heads carrying the tools for truing up the regular ways are now fastened to the end of the table, the machine is started up, and the ways are trued up from the center to one end. The heads are then placed on the opposite end of the table, and the remaining portions of the ways are finished. Then the sliding surfaces of the table itself are planed up by placing the same truing heads in pockets provided in the ways in the bed, near the center of the machine. The accuracy of the finished work depends, of course, entirely upon the accuracy of the auxiliary ways, and these are therefore finished on the completed planer with great care.

Fast Traverse and Cross-rail Operating Mechanism.

A motor for the fast traverse of the heads is shown at the end of the cross-slide in Fig. 10. The reversing is accomplished through friction clutches, and, in order to prevent the throwing in of the fast traverse and the feed mechanism at the same time, a special safety device is provided. The motor for operating the cross-rail is placed at the top of the right-hand upright, as shown in Figs. 1 and 10. This motor is connected at all times to the elevating screws, and is started, stopped, and reversed from the switchboard. The elevating screws for the cross-rail are firmly held at the top and bottom, and the cross-rail nuts are placed in square pockets against shoulders. It is expected that this will take care, satisfactorily, of the thrust of the slotter bars when in operation, but provisions have been made so that the cross rail can be firmly clamped to the uprights in case any trouble from loosening should be experienced.

FORMULAS FOR CIRCULAR FORMING TOOLS.

When laying out circular forming tools, such as shown in Fig. 1, the cutting edge, as is well known, must be located a certain amount below the horizontal center line of the tool, in order to provide for sufficient clearance for the cut. On account of this, the actual differences of diameters in the piece of work to be formed cannot be directly copied in the forming tool. The distance A in the piece to be formed must

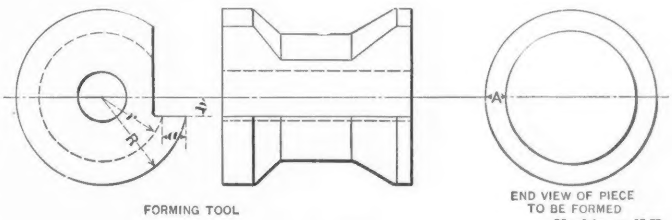


Fig. 1.

equal the distance a on the forming tool, but as this latter distance is measured in a plane a certain distance b below the horizontal plane through the center of the forming tool, it is evident that the differences of diameters in the tool and the piece to be formed are not the same. A general formula may, however, be deduced by use of elementary geometry by means of which various diameters of the forming tool may be determined if the largest (or smallest) diameter of the tool is known, the amount that the cutting edge is below the center, and, of course, the diameters of the piece to be formed.

If R = the largest radius of the tool,
a = difference in radii of steps, and
b = amount cutting edge is below center,
then, if r be the radius looked for,

$$r = \sqrt{(\sqrt{R^2 - b^2} - a)^2 + b^2}$$

If the smaller radius r is given and the larger radius R sought, the formula takes the form:

$$R = \sqrt{(\sqrt{r^2 - b^2} + a)^2 + b^2}$$

Suppose, for an example, that a tool is to be made to form the piece in Fig. 2. Assume that the largest diameter of the tool is to be 3 inches, and that the cutting edge is to be ¼ inch below the center of the tool. Then the diameter next smaller to 3 inches is found from the formulas given by inserting the given values: $R = 1\frac{1}{2}$ inch, $b = \frac{1}{4}$ inch, and $a = \frac{1}{4}$ inch (half the difference between 4 and 3½ inches; see Fig. 2).

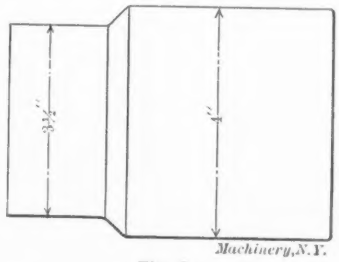


Fig. 2.

Then

$$r = \sqrt{(\sqrt{(1\tfrac{1}{2})^2 - (\tfrac{1}{4})^2} - \tfrac{1}{4})^2 + (\tfrac{1}{4})^2} = \sqrt{(\sqrt{\tfrac{35}{16}} - \tfrac{1}{4})^2 + \tfrac{1}{16}} = \frac{5.017}{4}$$

$= 1.254$ inch.

While the formula looks complicated, by means of a table of squares the calculations are easily simplified and can be carried out in three or four minutes. The value r being 1.254 inch, the diameter to make the smaller step of the forming tool will be 2.508 inches, instead of 2½ inches exact, as would have been the case if the cutting edge had been on the center line.

* * *

The usual plan followed in making punches and dies is to harden the die, leaving the punch soft. The punch is generally finished to exact shape and size by shearing it into the die. An alternative practice which is advocated by some die-makers for such work as armature laminations, is to make the punch hard and the die soft, it being argued that the wear of the soft part, which then is the die, does not tend to raise a burr on the punchings. When the punch is left soft it leaves a burr on the work as it wears, which is very objectionable in the case of armature laminations. If the die is made soft it is not necessary to frequently peen the punch and shear it into the die to overcome the burring referred to.

THE ELASTIC LIMIT AND THE TESTING OF MATERIALS.

H. GANSSLEN.*

The following article is intended to explain in a popular way a subject on which manufacturers, mechanics, and others interested purely in what is called the practical side of the machinery business, judging from my experience, are greatly in need of some definite information. It is not meant to contain any information for specialists on testing materials, although even the specialist may find some valuable data in it, as the diagrams and the figures given are the results of actual tests made by the writer when in charge of the testing department of the E. W. Bliss Co., of Brooklyn, N. Y., and also while assistant to Prof. Bach, the well-known German authority on testing and strength of materials.

The term "elastic limit" has lately become quite popular around the factories and in the sales offices, but I have met very few people who have a clear idea of what the term really means, and of what practical use its application is. I cannot help but cite a case illustrating this lack of familiarity with the subject, even among people who ought to know. A few years ago I made some commercial tests of a special high-grade steel in the presence of the representatives of the two contracting parties and of the maker of the testing machine. The latter was asked by a venerable looking official: "What is elastic limit, anyhow?" After he got half through with what promised to be a fairly good definition of the term, the old gentleman interrupted him with: "You may be all right, but I don't know what you are talking about." There was a complete silence, and I doubt whether anyone present at that time knows any more about that mysterious thing, elastic limit, to-day than they knew then.

Method of Testing Materials.

By means of the following diagrams, the writer will try to illustrate what happens to a round test bar of steel under a load P, this load starting at zero and increasing by the bars

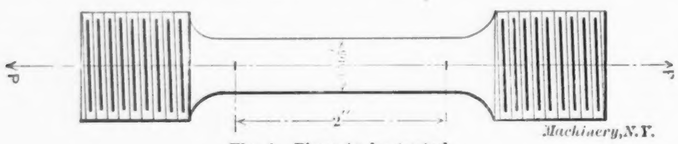


Fig. 1. Piece to be tested.

being stretched with a uniform speed of the machine, say 3/16 inch per minute, until the bar breaks. Fig. 1 shows the shape and dimensions of a bar which is, incidentally, the same as prescribed by the United States Government for certain purposes. The threads on both ends fit the two chucks on a testing machine, the upper one of which is stationary and a part of the machine frame, while the lower one is part of a cross-head moved by gears and screws which exert a tension on the test bar. Besides these parts, the machine consists of a system of levers, similar to the ones on scales, enabling the operator to see the load at any time. The diameter, 0.505 inch, of the test bar gives the latter a sectional area of 1/5 square inch, over which the total load is assumed to be equally distributed. The load on any test bar is, for the sake of comparison, usually expressed in pounds per square inch of its area, and is then called a unit-load. The resistance of the material is, of course, the same as the external force or load. A unit load of 1,000 pounds (external) causes a unit-stress of 1,000 pounds (internal), which holds the equilibrium. A thousand pound load on our test bar, for instance, would, therefore, cause $5 \times 1,000 = 5,000$ pounds unit-stress. After putting a load of 1,000 pounds on the bar, the length under observation, which was originally 2 inches, will then undoubtedly be greater, say 2.0003 inches. The quantity 0.0003 inch is called the elongation at the load of 1,000 pounds. This elongation for 1-inch length would be $0.0003 \div 2 = 0.00015$ inch, and this we call unit-elongation. It is generally assumed that all parts of the 2 inches of the bar under observation take an equal share in this elongation, or, in other words, the elongation is supposed to be equally distributed over the entire length. Elongations are measured with very fine instruments, extensometers, which are working either on optical or mechanical principles. It is a fundamental law that every stress should be accompanied by a deformation, however small it may be, and whatever material we may test, whether steel, iron, copper, brass, stone, concrete, wood, glass, etc., this holds true, but the amount of the deformation of these materials differs very much under the same stresses.

Fig. 2 shows a diagram giving the unit-stresses and the respective unit-elongations of a piece of high-grade steel of

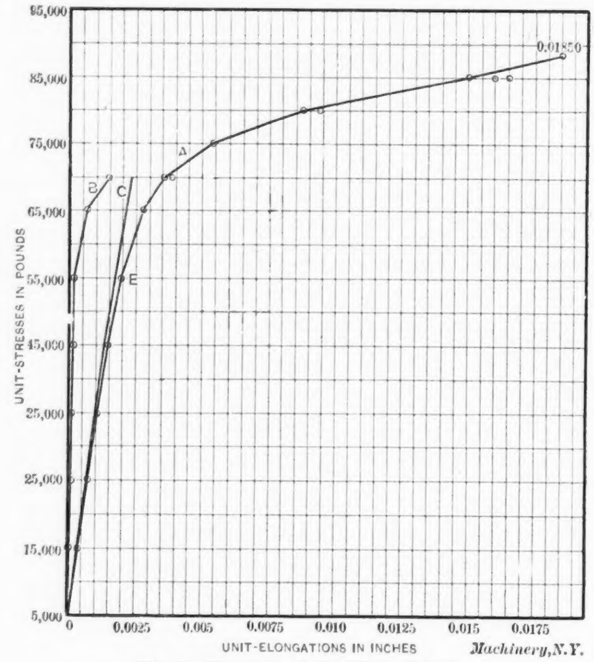


Fig. 2. Diagram of First Part of Test.

the dimensions given in Fig. 1, which I tested some time ago. Curve A gives the *total unit-elongation*, as per the accompanying table, B the *permanent sets*, and C is the curve of the *elastic unit-elongations*, which was arrived at by deducting B from A. The table gives also the actual record of the test, the load, as indicated on the beam of the testing machine and the corresponding extensometer readings. As seen from the table, a small initial unit-load of 5,000 pounds per square inch was put on the bar, to have all the parts of the testing machine and extensometer in tension. After gradually increasing the unit load to 45,000 pounds, there was a total unit-elongation of 0.00145 inch. At this point the load was reduced to the initial one, and there was noticeable a slight permanent set of 0.00015 inch. The difference of 0.00145 and 0.00015, equaling 0.00130 inch, is called the elastic unit-elongation, as it disappears completely after taking the

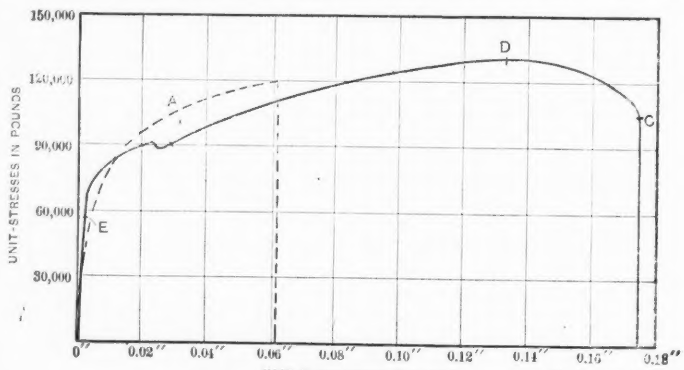


Fig. 3. Diagram of Complete Test.

load off. The permanent set at that point and up as far as 55,000 pounds unit-stress (point E, Fig. 2), is so small as to be almost negligible. It is due to some extent to the natural error in reading the instrument. Above point E the curves of the total and permanent elongations deviate from the hitherto straight line, *i. e.*, the permanent set of the material becomes now more marked. The stress 55,000 pounds is therefore called the *elastic limit* of this particular material, as any higher stress leaves a set in the material after releasing it, or, in other words, stretches the material beyond its elasticity. This test was continued up to a stress of

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70,000 pounds, and then the bar was unloaded down to the initial 5,000 pounds per square inch; the permanent set at that point was found to be 0.0029 inch. Above 70,000 pounds stress the total elongations only were measured, as the elastic elongations are of no further practical value, the material being stretched beyond the range which is of importance for commercial engineering purposes. In Fig. 3 are, however, given the diagrams up to the point of fracture, from which we are still very far away. The material beyond point *E* goes into a "flowing" state, as small increases in the load produce now much larger elongations than was the case below *E*. Furthermore, the bar stretches without

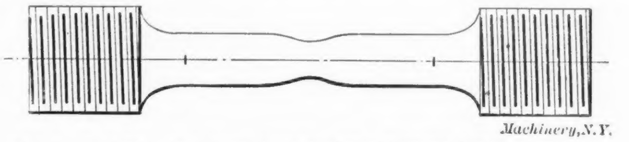

Fig. 4. Test Bar after having reached the Ultimate Limit of Strength.

increasing the load, as shown by the figures in the table, and also in Fig. 2, at 80,000 and 85,000 pounds stress. Time now becomes a factor. The different elongations given for each of the two loads were recorded within small fractions of a minute. At 88,500 pounds stress per square inch the extensometer was removed, as it had reached the limit of its capacity, and because the elongations could now be measured near enough for the purpose with a pair of dividers. A stress of 93,000 pounds brings us to the most remarkable point of the whole test. The material, without any outward signs visible to the naked eye, seems to utterly collapse; it stretches from

RECORD OF TEST ON STEEL BAR.

Load in pounds.	Stresses, Pounds per square inch.	Elongations, inch.	Unit Elongations, inch.
1,000	5,000	0	0
3,000	15,000	0.0007	0.00035
5,000	25,000	0.0014	0.00070
7,000	35,000	0.0021	0.00105
9,000	45,000	0.0029	0.00145
1,000	5,000	0.0003	0.00015
9,000	45,000	0.0029	0.00145
11,000	55,000	0.0038	0.00190
13,000	65,000	0.0056	0.00280
14,000	70,000	0.0072	0.00360
1,000	5,000	0.0029	0.00145
14,000	70,000	0.0075	0.00375
15,000	75,000	0.0108	0.00540
16,000	80,000	0.0173	0.00865
16,000	80,000	0.0186	0.00930
17,000	85,000	0.0300	0.01500
17,000	85,000	0.0320	0.01600
17,000	85,000	0.0330	0.01650
17,000	85,000	0.0370	0.01850

0.023 inch to 0.032 inch per inch length, and the load which the bar was able to stand in the meantime dropped down to 90,000 pounds, until it reached 93,000 pounds again at 0.032 inch elongation. The stresses 93,000 and 90,000 pounds per square inch are called the *upper* and *lower yield-points*, respectively. In practical life they are frequently, but wrongly, called elastic limit. After this stage the material revives again, and the load can be materially increased, the elongations becoming greater and greater for the same increments in load. The maximum load this particular test bar stood was 131,500 pounds per square inch. This is called the *ultimate strength* of this material. After this point was

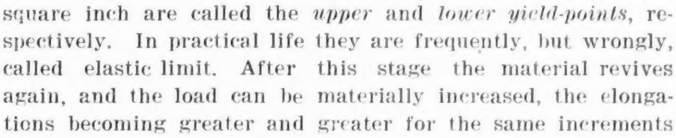

Fig. 5. Diagram of Test on Copper.

passed, the test bar was getting thinner at one point, it contracted more and more, and the load which the material stood became smaller and smaller, as the descending part of the curve, Fig. 3, shows, until the bar broke at point *C*. Fig. 4 shows the test bar after point *D*, Fig. 3, was passed, *i. e.*, after the local contraction became plainly visible. The elongation in 2-inch length, measured after the fracture, was 0.354-inch, or $\frac{0.354}{2} \times 100 = 17.7$ per cent. The smallest diameter at the place of contraction measured after the fracture was 0.380 inch, which gives a reduction of area, or contraction, of 43.4 per cent.

By planimetering the area of the diagram, Fig. 3, it is possible to find the work in foot-pounds necessary to tear the material apart. This is sometimes called *resilience*. The amount of work necessary to stretch the bar up to its elastic limit is called the *elastic resilience*, and it can be reclaimed. It is not intended here to further consider this subject, except to show by curve *A*, in Fig. 3, how little work it took to break a similar piece of steel heat treated in a different way. The test piece has almost the same tensile strength (120,000 pounds per square inch) but only 6.3 per cent elongation and 14.4 per cent contraction. One shock will lead to its fracture, where it would require several shocks of the same kind to break the material described above. The ultimate strength of a material alone is, therefore, no criterion as to its fitness for any particular purpose. It will be noticed from the diagram that there is no yield-point perceptible in this material.

It may be in place here to also explain in a few words a term which is often used—coefficient of elasticity. At 55,000 pounds stress (Fig. 2) there is an elastic unit elongation of 0.0018 inch. The coefficient of elasticity is, therefore $\frac{0.0018}{55,000} = \frac{1}{30,500,000}$. The modulus of elasticity is the value 30,500,000. The coefficient of elasticity is nothing else than the elastic unit-elongation per one pound stress. It is a measure of the

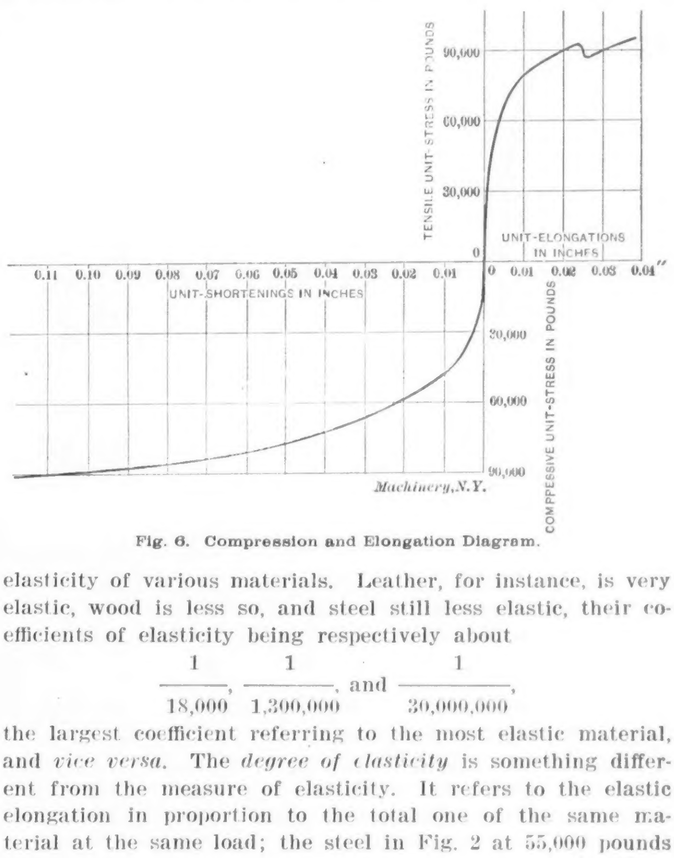

Fig. 6. Compression and Elongation Diagram.

elasticity of various materials. Leather, for instance, is very elastic, wood is less so, and steel still less elastic, their coefficients of elasticity being respectively about

$$\frac{1}{18,000}, \frac{1}{1,300,000}, \text{ and } \frac{1}{30,000,000},$$

the largest coefficient referring to the most elastic material, and *vice versa*. The *degree of elasticity* is something different from the measure of elasticity. It refers to the elastic elongation in proportion to the total one of the same material at the same load; the steel in Fig. 2 at 55,000 pounds stress, for instance, would show this ratio to be $\frac{0.0018}{0.0019} = 0.95$. The nearer this ratio comes to 1 the more perfect the elasticity is, or, in other words, any material showing no set at a certain load, after same is removed, is called perfectly elastic at that load.

Concluding Remarks.

As to the practical application of the term elastic limit, the buyer of materials, especially of steel with a specified elastic limit, should realize that it is very necessary to state what he means by elastic limit, it having become rather customary to use the term wrongly in place of yield-point. It is possible to obtain steel for about 3½ cents per pound, which will, if properly treated, have a yield-point of probably 90,000 pounds per square inch, besides having quite an appreciable elongation, say, 15 per cent. The buyer might, how-

ever, be very disappointed if he would try to treat it so as to obtain an actual elastic limit of 90,000 pounds per square inch, the term this time being used in its correct meaning as given in the above definition, which is generally accepted by engineers. Another factor which makes the word elastic limit a rather uncertain and vague term, is the speed at which the test bar is being stretched during the test. For the test piece shown in Fig. 1, the United States Government in certain specifications requires a speed of the testing machine, that is, an elongation of the test bar, of from ½ inch to 3 inches per minute. Aside from the large margin it leaves the operator of the machine, even the smallest of specified speeds appears much too large for such a short test bar, especially where high-grade steel is used, inasmuch as the load comes on very suddenly, almost like a shock. Under such conditions it takes a very skilled operator to determine the yield-point with any degree of accuracy. The yield-point, as well as the ultimate strength, is greater under such high speed than under slower ones. This difference has been found to amount to as much as 15 per cent. (See C. Bach, Elasticität und Festigkeit, 1898, page 129.) Government inspectors are very strict, as they ought to be, with regard to the margin below and above the elastic limit required. It seems to be an urgent necessity to introduce proper rules regarding the speed of tests, and to the writer it appears that a certain maximum increment of stress per minute for each class of material would be a better way of putting it than the one mentioned above, as this would eliminate the proportions of different test bars.

Fig. 5 shows a diagram of a test of copper under tension, having no yield-point at all; the same applies to the particular steel shown in curve *A*, Fig. 3. Not all materials possess a yield-point. As a criterion as to what stress we should then allow on any part of a machine or structure, we will have to use the curve of the total elongations and permanent sets. For special purposes, as for instance, embossing press work, we are only interested in the curve of the permanent sets of the material to be embossed. Without going further into this subject at this time, there is shown, in Fig. 6, a curve of the permanent sets of a Bessemer steel cylinder under compression (Transactions of A. S. M. E., 1906) as compared with the steel in Figs. 2 and 3 under tension. Whereas, the yield-point of the tension curve is plainly marked, it hardly is marked at all in the compression curve. There we might consider it as being at from 50,000 to 60,000 pounds per square inch, and the material's range of usefulness for embossing lies above these points, below which its applicability for ordinary engineering purposes has already ceased.

* * *

It is less than a decade ago since the first turbine was sold in the American market, but there are to-day about 700 in use throughout the country, aggregating a total capacity of approximately 1,000,000 kilowatts or about 1,350,000 horse-power. This great demand for a new prime mover is, of course, easily explained by the advantages the turbine has over the reciprocating steam engine. An interesting test was conducted recently by the engineers of the New York Edison Company at the Waterside Station near 30th Street, which developed facts hitherto unattained by any steam prime mover in this country. The unit under test was a Westinghouse turbine of 10,000 horse-power capacity. It had been sold under a steam consumption guarantee of 15.9 pounds of steam per kilowatt hour, but the test recorded the phenomenally low steam consumption of a shade less than 14.9 pounds per kilowatt hour. (See MACHINERY, November, 1907, engineering edition.) Apart from the fact that this result gained a bonus for the Westinghouse turbine of over $25,000, it is of interest to all users of steam engines as an illustration of the lowest record for steam consumption which has ever been recorded by a stationary steam engine. This steam consumption figures less than 1½ pound of coal per kilowatt hour. It should be mentioned that the steam supply was superheated, which fact, of course, materially reduces the steam consumption without a corresponding reduction in consumption of heat units.

NEW SHOP OF THE MUELLER MACHINE TOOL COMPANY.

Improvements and new buildings have been the order of the day with most machine tool firms during the last two or three years, and we have from time to time recorded some new shop constructions in our columns. In this issue we show two half-tones of the interior appearance, and a number of line cuts of the plan and elevations, of the new shop of The Mueller Machine Tool Co., Cincinnati, Ohio. This shop has been completed during the year just past, and the machinery and equipment has just been installed. The general arrangement of this shop, as well as some of its commendable constructional features, may prove both of interest and of suggestive value to persons who either directly or indirectly may be connected with the planning of new shops or additions to present plants.

The feature which is most strikingly apparent when one examines the two half-tones showing the interior of the shop is the excellent light afforded by the numerous and large windows as well as by the skylights above. To persons acquainted with shop photography, a good idea of the excellence of the light is afforded by the statement that the photographs from which the half-tones shown were reproduced were taken with a 5 seconds exposure, F-16 stop, at 12:15 P. M. The main shop is a one-story building, but the rear portion of the shop is provided with a basement which, in fact, adds another story for half the length of the shop. This basement has the same advantages of good light from the outside as the main floor, on account of the difference of the ground level at the front and rear of the shop, the slope at the rear end amounting to about 12 feet. In this basement all the unfinished stock and other raw materials are stored. The floor is made of concrete, and wide doors are provided from the yard so that a wagon may drive right into the basement for unloading, a traveling crane being provided for facilitating this work. In the basement is also installed a Stewart combination hot blast furnace, and the forging shop is located here. All hardening, case-hardening, etc., also takes place in this portion of the shop.

The floor level of the main shop is three feet above the street level, which makes it very convenient for loading and unloading machines to and from trucks, the wagons or drays being backed into the building at the drive-way shown in the lower part of the plan view, Fig. 3. The opening into the street from this drive-way is provided with steel rolling shutters which can, without much trouble, be closed, and thus, during cold weather, as soon as the team has been backed into the drive-way, the shutters may be closed during the loading and unloading, thereby retaining the heat in the shop. All machine tools are installed on the rear portion of this floor, one side and the middle of the building being almost entirely given up to the machine tools themselves, while the erection and bench work is done on the other side of the building. In the front of the building are the offices, drawing-room and the shop washroom, which latter is provided with individual lockers for the men. A special stock-room for finished material is located behind the drawing-room, or rather, between the drawing-room and the shop. The main entrance to the building opens into a vestibule from which one may enter either the office to the right or the shop directly in front. At the rear of the building the freight elevator is installed for carrying material between the basement and the main floor. The platform of this elevator is 6 feet by 9 feet, and it has a carrying capacity of 5 tons. The stairs leading down to the basement are located immediately beside the elevator. Near the elevator, also at the rear of the building, is shown a steel chute through which all the chips from the machine operations are dropped. The chute leads to a concrete bin outside of the building, in which the chips are stored until disposed of. The building is steam heated, the steam being generated by an "American" low-pressure boiler. All the benches along the wall are three inches from the brick wall, in order to permit the pipes to be carried near the wall and keep the direct radiation of the heat away from the men working at the benches.

Of the general specifications for the shop the following data

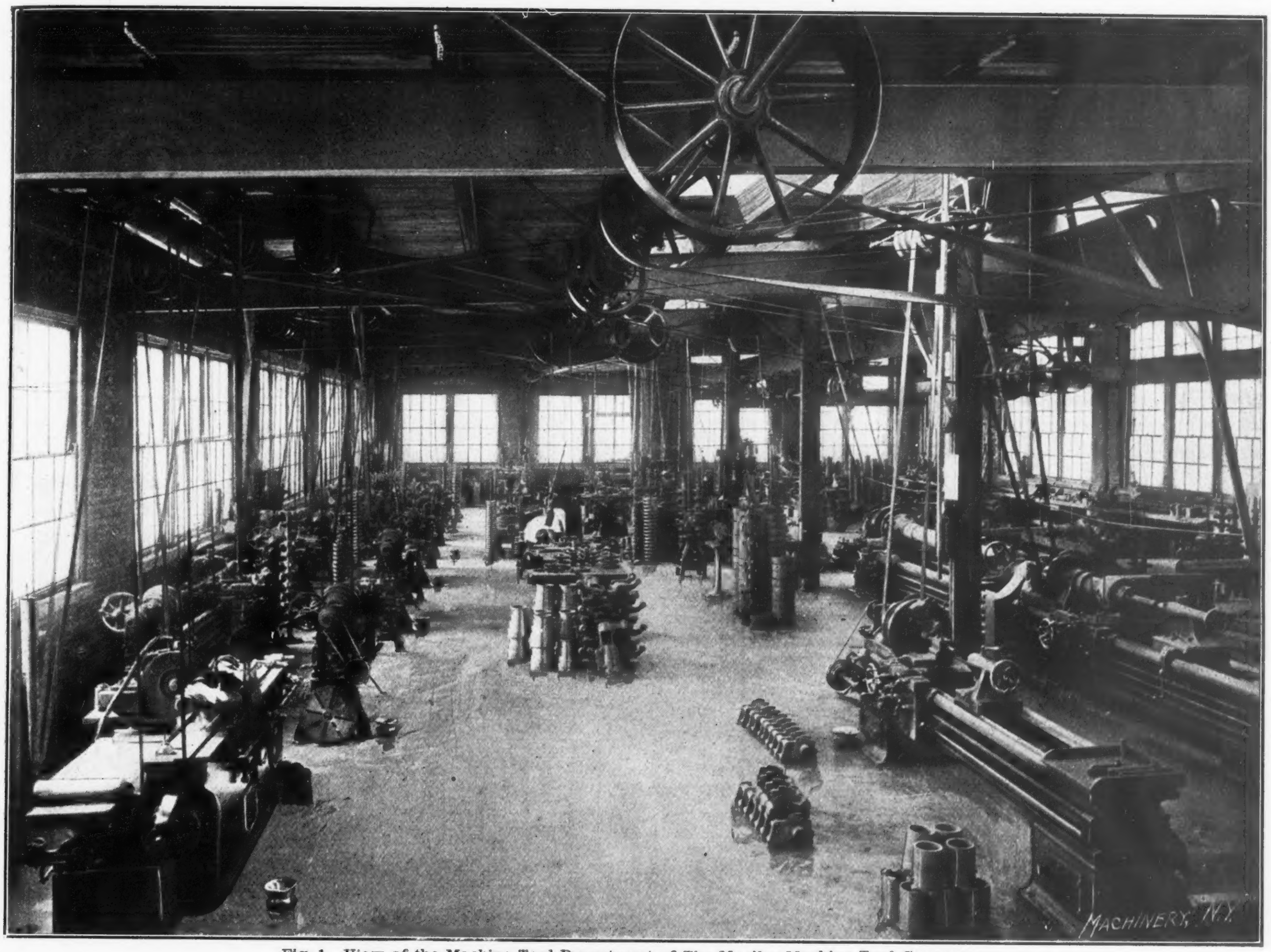

Fig. 1. View of the Machine Tool Department of The Mueller Machine Tool Company.

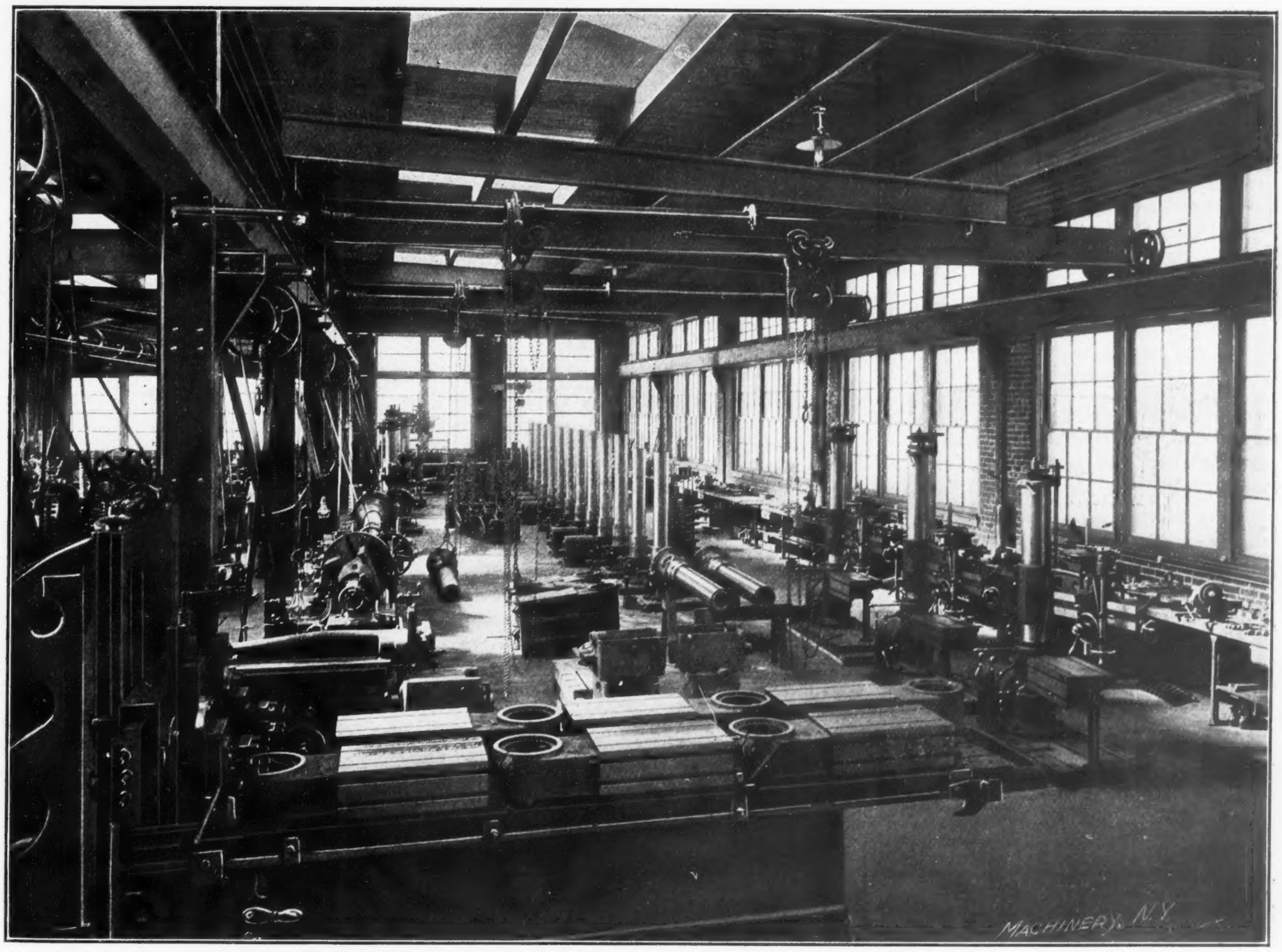

Fig. 2. View of the Erecting Department of The Mueller Machine Tool Company.

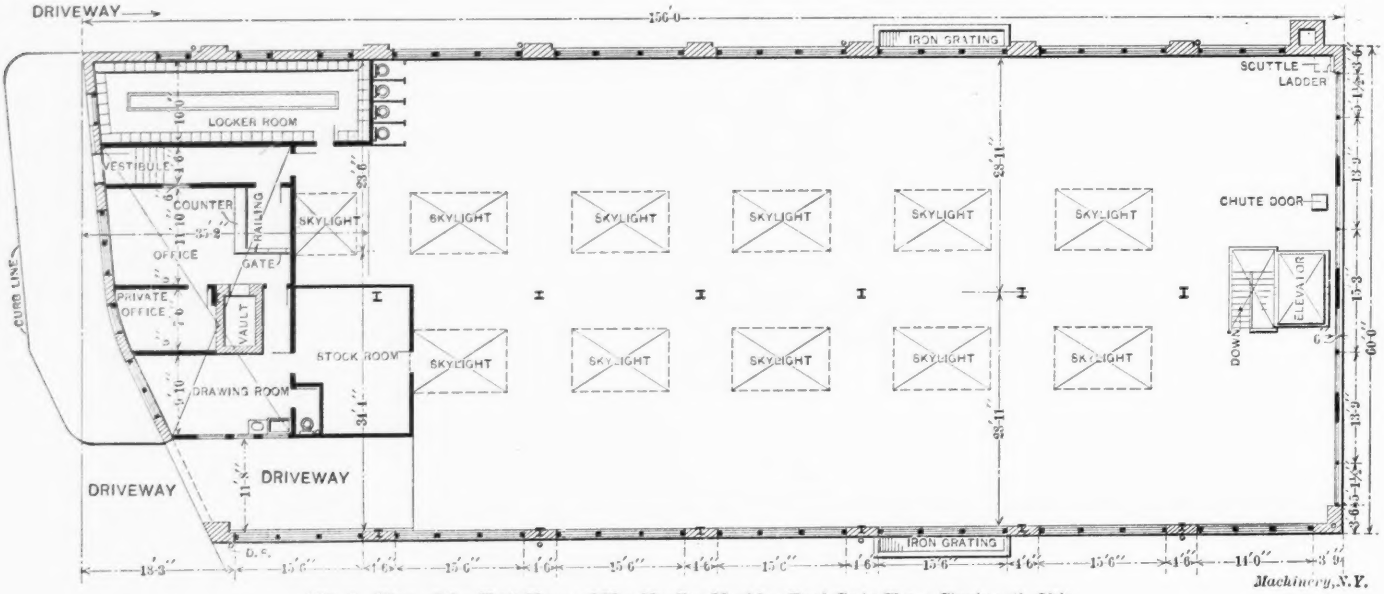

Fig. 3. Plan of the First Floor of The Mueller Machine Tool Co.'s Shop, Cincinnati, Ohio.

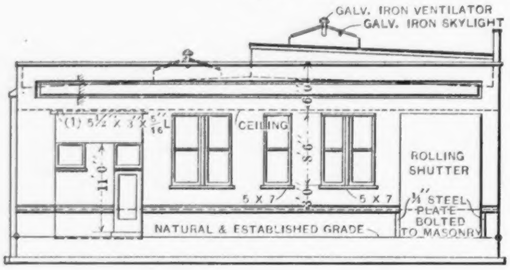

Fig. 4. Front Elevation.

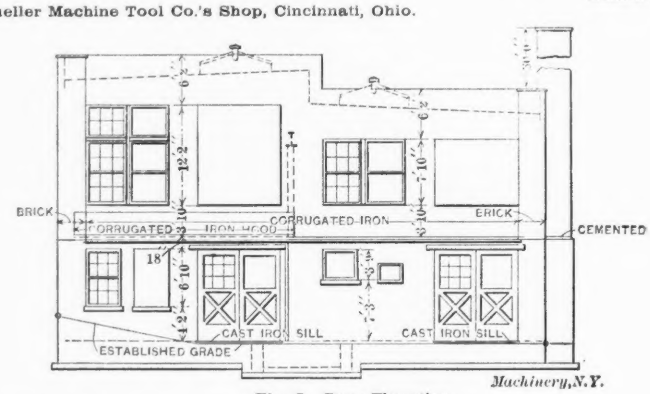

Fig. 5. Rear Elevation.

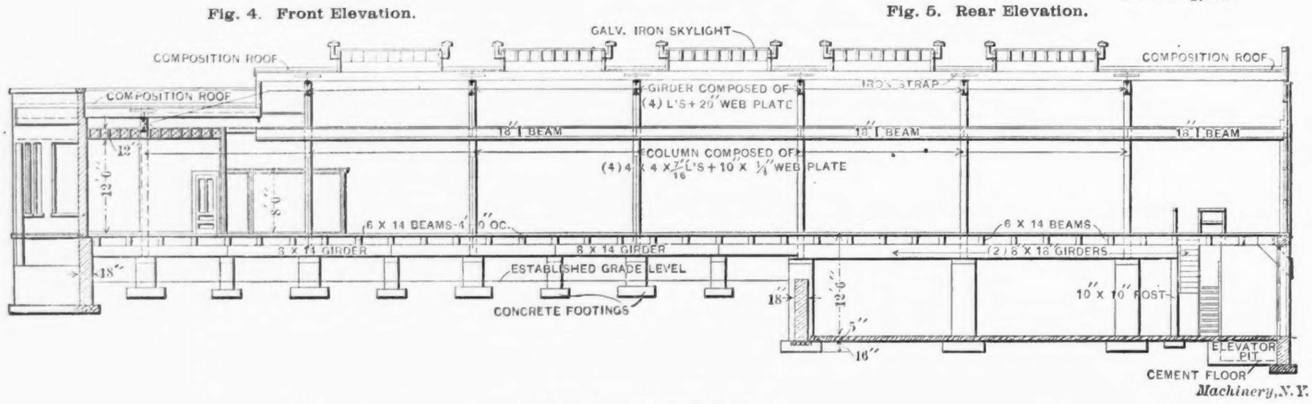

Fig. 6. Longitudinal Section.

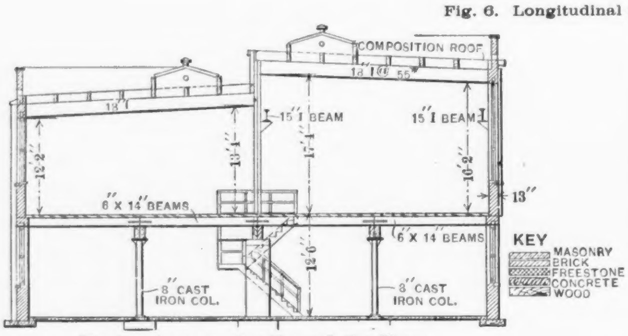

Fig. 7. Cross-section through the Shop.

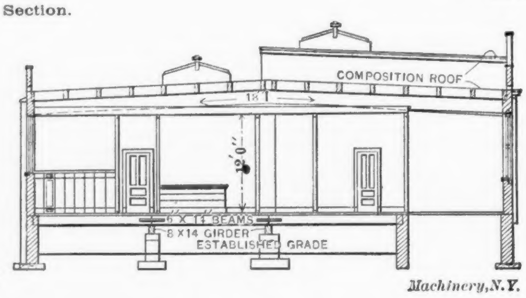

Fig. 8. Cross-section through the Office Partitions.

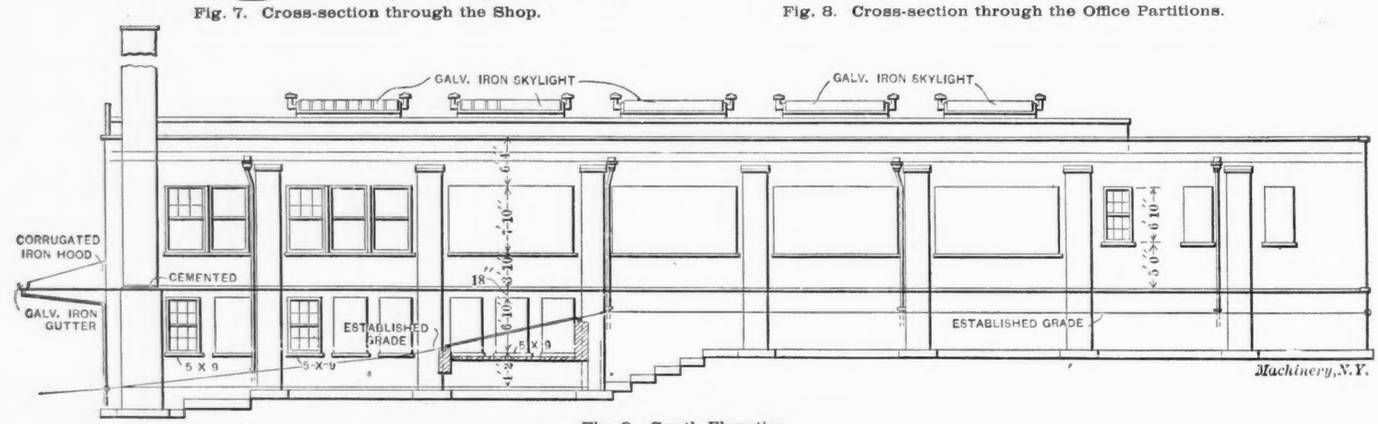

Fig. 9. South Elevation.

may be of interest and of suggestive value. The floor on the upper or main story in the shop is laid diagonally with 2¼-inch tongued-and-grooved pine lumber, and has a top layer, laid diagonally at an opposite angle, of ⅞-inch tongued-and-grooved maple lumber. The steel columns through the center of the building are placed 20 feet apart, and carry 18-inch I-beams which in turn support the roof. The erecting side of the building has a clear height of 16 feet, it being possible to thus obtain a 12-foot lift with the cranes installed, each of which has a capacity of 4 tons. Besides the traveling cranes, there is at the rear wall of the building a 15-foot jib crane, intended to be used by the vise hands when handling parts of machines for assembling, such as heads, arms, tables, etc. The side of the shop, which is exclusively given up to the installation of machine tools, is 12 feet high, this height being the most convenient for the installation of counter-shaft and belting. The skylights, previously mentioned, are ten in number, each measuring 6 feet by 10 feet, projected area. By means of these skylights the middle of the shop is fully as well lighted as are the sides near the windows, and the light over the whole shop is diffused in the most satisfactory manner. The windows all come up close to the ceiling, and those on the north and south sides have transoms for ventilating purposes. Ventilators are also placed one at each end of each skylight.

The line-shaft is motor driven, a Bullock motor and a Renold silent chain constituting the drive. The motor, as will be seen from Fig. 1, is attached to the ceiling, as usual in this kind of installation. The line-shaft is provided with two Dodge clutches, so that sections of the line-shaft can be thrown out if required. All the line-shaft hangers, as well as the counter-shaft hangers, are supported by angle plates bolted directly to the I-beams, thus doing away with all wooden hanger planks. All large machine tools are placed on concrete foundations, while the smaller ones are simply bolted to the floor.

The general construction and equipment of the shop shows care and forethought, everything having been arranged so as to be advantageous for economical manufacturing as well as convenient for the men, and the shop constitutes a good example of moderate sized shop construction adapted to machine tool manufacture.

* * *

HOW A LARGE CASTING WAS PLANED ON A SMALL PLANER.

The photograph for the half-tone cut, Fig. 1, was sent to MACHINERY by Mr. L. T. Wilmarth, of the Wilmarth & Morman Co., Grand Rapids, Mich., to illustrate how a machine frame casting weighing about 1,200 pounds, and of very awkward shape, was planed on a 24 x 24 x 6-foot Woodward & Powell planer. While the method is not new it is of general interest to small shops which frequently are called upon to perform work that is beyond the ordinary capacity of their machine tools.

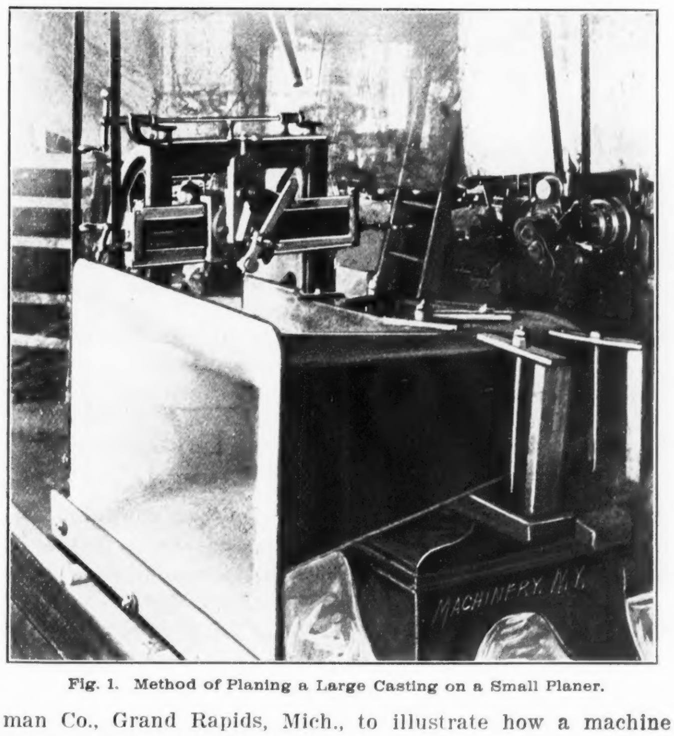

Fig. 1. Method of Planing a Large Casting on a Small Planer.

The surfaces to be planed were the two opposing faces of the jaws indicated by *A* and *B* in Fig. 2, being the upper and lower sides of the projecting arms at the outer ends. The upper surface was 7¼ inches long by 14 inches wide, and the lower surface was 9¼ inches long by 14 inches wide. The lengthwise dimensions are in a direction parallel to the direction of the arms. The casting was about 63½ inches long, measured from the end of the lower jaw to the perpendicular through the back edge of the base; the width was 30 inches measured on the base; and the height was approximately 58 inches.

Obviously the casting could not be planed on a 24 x 24-inch planer without allowing the base to over-hang the table. The over-hang was about 3 feet and it was carried on a framework partly shown in Fig. 1. Its construction, however, was simple, consisting of three wooden blocks, 12 x 12 x 24 inches, laid on the floor at right angles to the planer bed, on which was placed a 9 x 12-inch x 12-foot timber lengthwise with the planer. This timber was strongly braced sideways to hold it in place, the braces being spiked to the timber and to the floor. A flat iron runway or track was screwed on top of the long timber and upon this track were placed two 2-inch rolls. On account of the unevenness and irregularity of the base, it was necessary to provide a straight-edge bearing for the casting which could be made parallel with the planer ways. This was made with two pieces of flat iron ½ x 4 inches x 4 feet long bolted or clamped to the lower edge of the base.

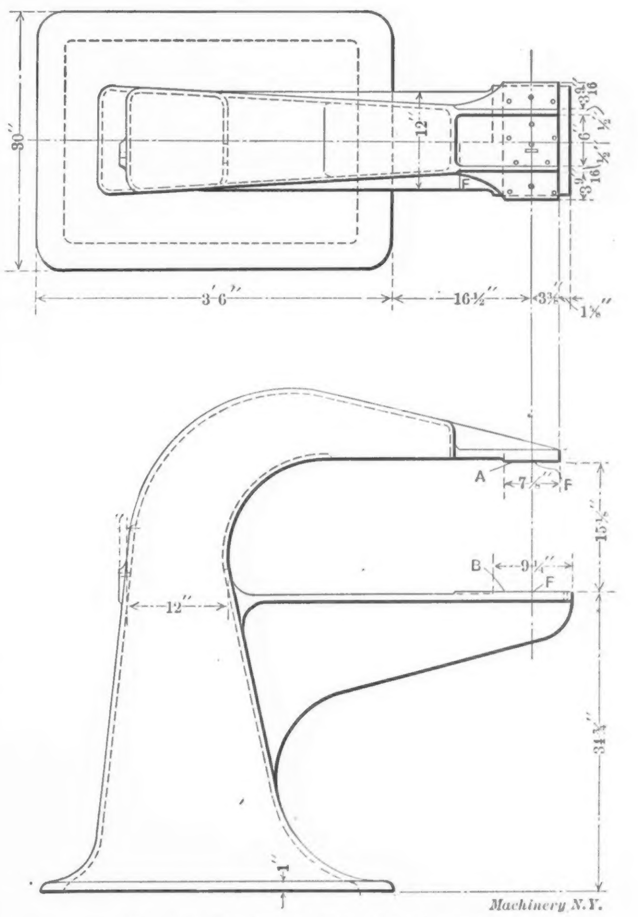

Fig. 2. Elevation and Plan of the Casting.

The time required for getting the outfit together and assembled, and the planing of the first casting was 28 hours; the time required for the second casting was 18 hours, but before the job of 20 castings was completed, the time was reduced to about 12 hours each.

* * *

At the end of 1906 the electric railways in the United States aggregated 36,212 miles. The equipment included 66,206 electric motor cars. Canada had at the same time 1,073 miles of electric railway.

MILLING OPERATIONS ON VISE PARTS.

JOHN EDGAR.*

Most of us are familiar with the flat machine vise, and we put a great deal of dependence upon its being square for the quality and accuracy of a great deal of our machine work, especially that done on the milling machine. The base must be parallel and square with the platen of the machine. The jaws must stand perpendicular with the top of the table, and lie at right angles to, or parallel with, the direction of the feed when the vise is tongued to the table. In order to fulfill all these conditions, and yet have the price of the vise reasonable, considerable planning must be done. The job of machining the parts that require the greatest accuracy is usually done on the planer. These parts are—the base, the slide, and the jaws.

The illustrations that accompany this article show how the different phases of machining these parts are done on the milling machine, and they represent the practice of a New England concern. The vises are manufactured in lots of from one hundred to three hundred. Little rigging up was necessary, and, outside of the cutters, the work called for but little extra expense for equipment. The form of the pieces is such that they are easily held on the naked platen by ordinary means—clamps and bolts. With three exceptions, no fixtures were used. The cutter gangs were for the most part made up from cutters on hand, as in but few of the operations were special gangs necessary. Had the several operations been carried on for a larger number, or for smaller lots, more frequently the case would have been different and special gangs would have been necessary. Of course, in work of this kind, the use of special gangs set up on an arbor for each gang eliminates the necessity of making up a gang and grinding same for the job to be done. However, under the circumstances, the results were, and still are, very satisfactory.

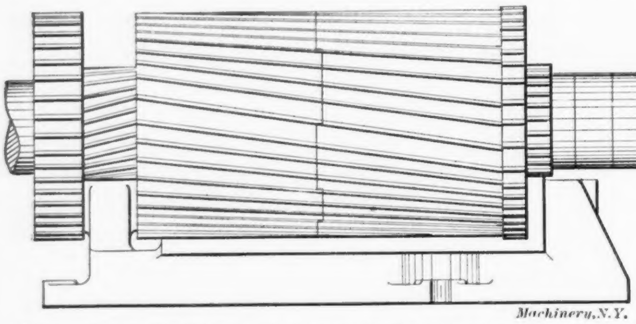

Fig. 1. Rough milling the Top of Base, Jaw Top, Jaw Backing and Screw Bosses with Gang Cutter.

First Operation—Milling the Top of Base, Jaw Top, Jaw Backing and Screw Bosses.

The first operation on the base is shown in Fig. 1. For this, a special gang of cutters was used, as the relation of the several cutters in the gang made it necessary. This operation is that of roughing off the top of the base, the front jaw, "backing," and the bosses for the screw support. The base casting has a spot on the top of the screw support lug that is machined off so that it comes level with the top of the front jaw backing. The object of having it so is to furnish a bearing on which to rest the base when machining the bottom. The large slabbing cutter in the center of the gang is made interlocking so that the proper distance over all may be preserved. One end of this cutter is made with side teeth for milling the boss. The front jaw backing is milled by a side milling cutter made a trifle larger than the large cutter that lies alongside of it, so that a trough the width of the cutter is cut across the top of the base. The excess in diameter of the narrow cutter over the other is small, being about 1/32 inch. The reason for this narrow trough will be explained later on.

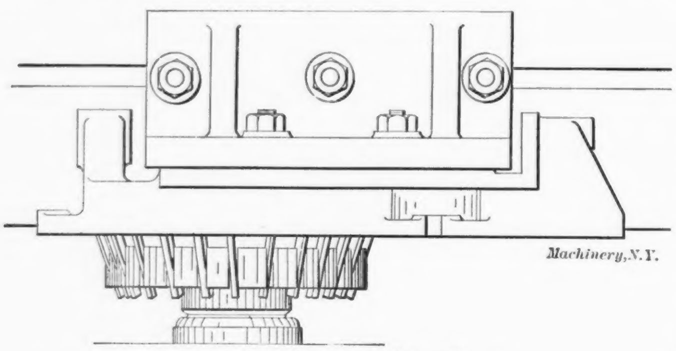

Fig. 2. Roughing Under Side of Base.

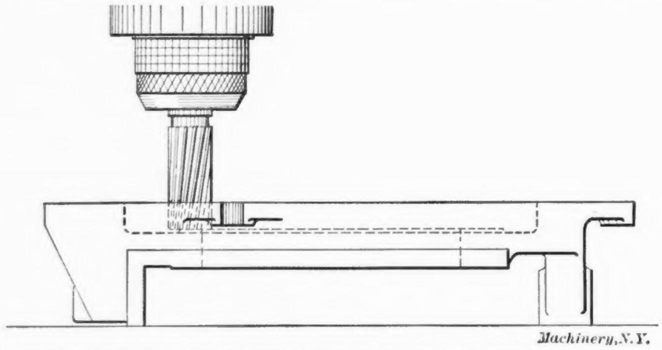

Fig. 3. Milling Under Strap Strips with End Mill, using Vertical Spindle Machine.

This operation was done on a heavy slabbing machine, and, while it was supposed to be a roughing cut, care was taken to have the finish fairly good, as the only surface re-milled was the top of the base. The pieces were strung on the platen, which was eight feet long, and were clamped as is ordinarily done on a planer. The feed was about three inches a minute, using high-speed cutters. The chip was light, only enough stock being left on the pattern to insure the removal of all the scale.

Second Operation—Roughing Under Side of Base.

The second operation was the machining of the under side of the base. This was accomplished on a horizontal milling machine, as shown in Fig. 2. The piece was held against an angle-iron bolted to the table. The reason for using the large inserted tooth cutter is that such a cutter will remove a greater amount of stock for the same expenditure of power, and will not strain the machine or holding devices so much as would a slabbing cutter. The face mill does not heat the work to the extent that a slabbing cutter does. The feed can be forced much faster than with the plain cutter, and the surface will not show the effects of such forcing. In all, a much better surface can be obtained with a mill of this kind on a broad surface than with the slabbing cutter. The feed, in this case, was forced to seven inches a minute, and the surface produced was all that could be desired. The feed was then dropped to five inches per minute, in order to save the cutter from repeated grindings. The bottom surface of the pattern for this piece is cut out considerably in the center, which, of course, counts for considerable in time of milling the resulting castings.

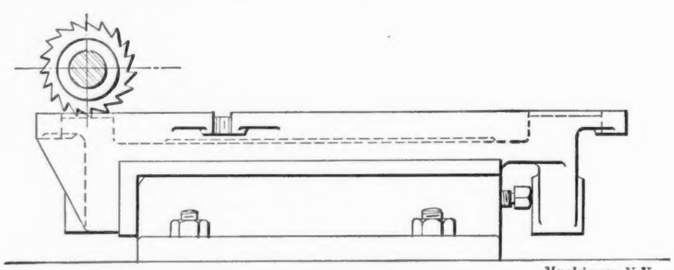

Fig. 4. "Tongueing" the Base.

Third Operation—Milling Under Strap Strips with End Mill.

For the next operation we find ourselves among the vertical millers. Here we have the base turned bottom upward, and an end mill machining the strips for the binder straps to ride on. There are two of these strips, one on each side of the center line. Each strip has a slot running nearly its full length, through which the screws pass. The advantage of the vertical milling machine on this operation is apparent. The cut is in full view, and the ease with which the cutter can be dropped in to the proper depth is easily imagined. This

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is the class of work for which the vertical spindle machine is peculiarly adapted. The proper depth for each piece is obtained by means of a gage block which is set under the micrometer "set" screw on the side of the head. The screw is then adjusted until the cutter just touches the bottom of the base. The block is then removed, and the head is lowered until the screw strikes the shelf.

The milling cutter is as small in diameter and as short as is convenient, so that a high speed and a fine "quick" feed may be obtained, leaving a surface tnat is easily scraped. In making it possible to obtain a good reasonable rate of speed, and still not overdo the matter.

Fifth Operation—Milling the Slots for the Movable Jaw.

The next operation, shown in Fig. 5, is also one that requires care, and on it depends the accuracy of alignment of the sliding jaw with the front jaw. This operation is that of milling the slots in the top of the base for guiding the

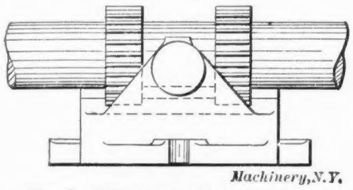

Fig. 5. Milling the Slots for the Movable Jaw.

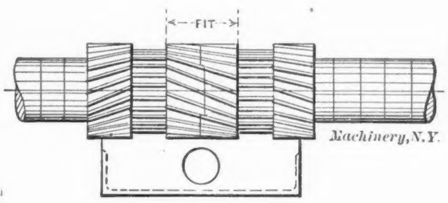

Fig. 8. Milling the Bottom Surface of the Slide and forming the Grooves.

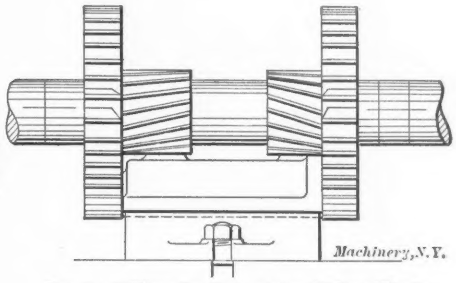

Fig. 9. Milling the Top of the Sliding Block.

this position, we use the spot spoken of in the early part of the article. The casting is clamped to the table by ordinary clamps and bolts.

Fourth Operation—Milling the Tongue Slots.

We now come to an operation, shown in Fig. 4, that requires a little care. This operation is milling the tongue slots—the most important feature of the flat vise, because should they be out of line or out of square the vise cannot be used with satisfactory results. The slots lengthwise of the base must be square with the front jaw, and the slots across the base

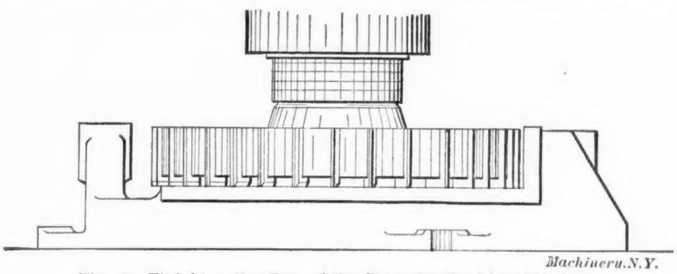

Fig. 6. Finishing the Top of the Base for the Movable Jaw.

must be parallel with that jaw and, of course, at right angles with the other slots. To procure good results in this case, the casting is set bottom up on a specially adapted block which is made perfectly square with the direction of the feed and parallel at one end with the axis of the cutter arbor. The top of the block, of course, must be parallel with the top of the table. The block is fitted with a set-screw which is brought to bear against the boss on the screw support lug, thus forcing the vise base home against the lining surface of the block. The casting is clamped to the block by means of

slide. In order to reduce the labor, the guides are made to fit on the inner side of each slot only, and a clearance is left on the outer side. The work, in this case, is clamped to the table, using tongues in the table slots to line it up. The base thus rests in a natural position, and the accuracy of alignment of the machine is the only feature upon which we place any dependence. Naturally the machine used for this work is previously tested for error. The statements regarding the cutters, in the last operation, apply with equal force here.

Sixth Operation—Finishing the Top of the Base for the Movable Jaw.

The work is again placed on the vertical milling machine, and, with a large inserted tooth face milling cutter, the top of the base is given the final cut. The operation is shown in Fig. 6. The cutter used in this operation is kept in good condition, exclusively for this work; it runs very true. The speed and feed of the cutter in this operation are rather above and below—respectively—that which would be considered correct for ordinary work, surface and finish being the main object of this cut. The teeth of the cutter are broad, and the clearance is small, insuring a smooth cut. This operation takes out any twist that may have been caused by releasing the scale on the bottom. For that reason the operation was delayed until this time.

It was for convenience in this operation that the trough was milled across the top of the base, as explained in connection with the first operation. The large face mill is set to depth so that it just scrapes the bottom of this trough, and also so as to clear the jaw backing as shown. It is obvious that should any attempt be made to mill both surfaces by a single operation, poor results would be obtained. The strain

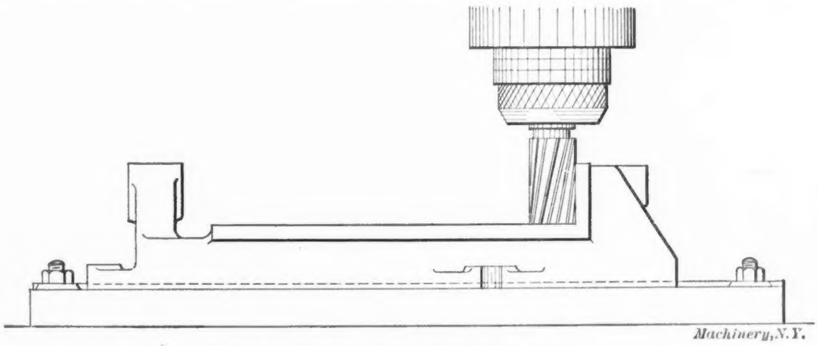

Fig. 7. Finishing the Jaw Backing Surface.

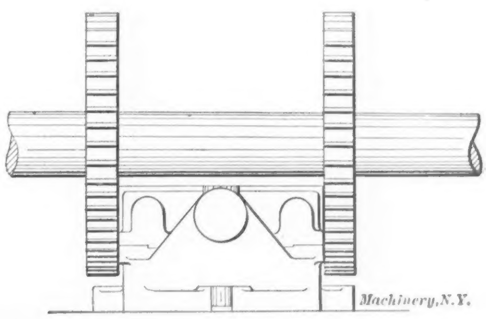

Fig. 10. Finishing the Sides of Base and Movable Jaw.

tap bolts through the slots in the casting and tapped into the block. A similar block is used to mill the cross slot, the only difference being in the location of the tongue.

The cutter used to slot the casting in this operation is a plain cutter with outside teeth. The slot is narrow, and, with the cutter properly hardened, little trouble is experienced from the corners of the teeth wearing excessively or breaking. A side milling cutter under such conditions would be under size after the first grinding. The cutter is made small in diameter, and is run at a correspondingly high speed, would cause chattering. By the method described, the top is milled independent of the vertical surface, and yet no corner is left that the milling cutter passes over. It also avoids the obvious difficulty of making a large cutter, screwed onto the spindle nose, run perfectly true on its periphery.

Seventh Operation—Finishing the Jaw Backing Surface.

More as a precaution than as a necessity, the operation shown in Fig. 7 is taken. The object is to straighten up the backing for the front jaw, and also to insure its being square

with the tongues, but here, again, we have to depend on the alignment of the machine. Probably it would be as well to omit this operation, for, when all conditions are looked into, the risk of throwing already accurate work out of square is great.

Eighth Operation—Milling the Bottom Surface of the Slide and Forming the Groove.

The operation of milling the bottom surface of the slide is shown in Fig. 8. This operation forms the tongues that fit the slots formed as shown in Fig. 5. The fit mentioned in dealing with that operation is indicated in Fig. 8. The cutters used in this operation are a special gang kept for the purpose. The block is clamped to the table by ordinary means.

Ninth Operation—Milling the Top of the Sliding Block.

The operation of milling the top of the sliding block is shown in Fig. 9. For this operation a casting was provided, having slots corresponding to the slots in the top of the vise base, at right angles to the tongue on the under side of this casting for lining same with the slots in the machine table. This course insures that the backing for the sliding jaw will be parallel with the line of the front jaws.

Tenth Operation—Finishing the Sides of the Base and Movable Jaws.

Fig. 10 shows the operation of milling the sides of the vise base and slide, the latter being put in position on the base and clamped to same. This operation mills the sides of both pieces so that they will be flush. This method was adopted rather than that of making special fixtures and special cutters for milling each piece separately.

As examples of operations wherein the milling machine has "butted" into the planer's territory, the above are well worth attention, and it is regretted that photographs were not available instead of the scanty line cuts. However, even these show the milling machine's superiority for such work to good advantage. They furnish, also, examples where equipment has in no way been a main point, and show that a large range of work can be handled with the ordinary means available. The various operations also show the different kinds of milling, and both styles of machines.

* * *

MOVING A SHOP.

B. HARDIE.*

It is sometimes necessary to move a manufacturing plant, particularly one renting quarters, from one place to another, either because the present place has to be vacated to give room for somebody who is willing to pay higher for the lease, or because cheaper and better quarters can be obtained elsewhere. This moving time is one of uncertainty of hours for all employed, and it makes the young apprentice swell with pride to have the superintendent approach him and ask him to work to-night, and he feels under moral obligation to tell every one he meets that "we are moving now, and I will have to work to-night."

At a time of such a shop moving, the most important thing is to have everything about the moving settled beforehand. The men in charge must determine upon exactly where every bench and machine should be placed in the new building in order to provide for the greatest convenience. The first thing to do is to obtain a drawing of the place, either from the owners renting it, or, if the place be bought, from the previous owners, or by having the draftsman sketch it up. In these plans particular attention should be paid to columns, height of ceiling, and location of cross beams, as well as the location of the windows. The location of the cross beams is very necessary in order to be able to arrange the counter-shafts correctly, and as for the columns, it is plainly in evidence that it would be inconvenient to have one of them located right in between the ways of a lathe. The same applies to the planning of closets in front of the windows, which mistake is often made in the drawing, if the windows are not plainly marked. When all these data are obtained, a plan should be laid out to as large scale as convenient. It is preferable to paste the paper on which this plan is laid out on the drawing board itself. Templets corresponding to the various machines in the shop should then be cut out of stiff cardboard, these templets being made as close to scale as possible of the actual floor space occupied by each machine. Particular attention should be paid in this instance to proper clearances, and the pasteboard pieces should be large enough to cover all projections or overhanging parts of the body of the machine, as these parts often will require larger space than do the columns or feet of the machine. Templets should also be made for benches, closets, and all other fixtures required to be installed. When this is done, the superintendent and the foremen in charge should arrange the templets as they consider best for carrying out the work in the most economical manner. When the templets have been arranged in a manner satisfactory to everybody concerned, thumb-tacks may be stuck to each templet as it is placed in position, and then another layout can be drawn up very easily, a tracing made, and blue-prints taken, so that everybody concerned in the moving can use one of these blue-prints as a guide.

One of the greatest troubles met with is the lining up of the shafting and the placing of the counter-shafts, which, if not properly done at first, will cause an endless amount of trouble later. A great deal of this difficulty is, of course, avoided in the more modern shops, where individual motor drive obviates this difficulty. Another very important point to be considered when all machines, benches and shafting are placed in position is the arrangement of the lighting fixtures, whether the lighting be by gas or electricity. This work should not be entered upon before everything else is in place, as one of the most important things in regard to light is that it be placed exactly where it will be needed the most, and this cannot very well be determined in the drafting room, but should be determined directly in the shop, when all the machines are in place.

Some shops will think it advisable to hire a few extra hands for the moving, but generally this is bad practice if it is possible to go along without extra movers. Hiring outside help for moving gives the men in the shop a chance to say that they "were not hired for moving," and may retard the work rather than facilitate it. Besides, in the case of moving it is a difficult matter to have too many men to look after, and the experienced old hands will probably know more about how to do the work properly than would extra hands. When everything is arranged right, each machine should be moved from its old place and put directly into its place in the new shop in the same order as it is moved. It is a great deal better to have each machine put in place at the time of moving than to have them all piled up and sorted out later, and if a man with a blue-print of the layout of the floor is appointed to superintend the placing of the machines in position as soon as arriving in the new place, it is an easy matter to attend to. The entire operation of moving can be more quickly accomplished if two crews are employed, one to take the machines from the floors and place them on the wagons at the old shop, and one to take them off the wagons and put them down in their place in the new shop. Thereby the time lost by the men traveling forth and back between the two shops is saved. If the moving is carried on along the lines indicated, and superintended in a proper manner, the interruption in business will not be noticed by outside customers, as each machine can be kept in running order up to the time of its removal, and be put in running order within a short time of its being placed in the new shop.

In the case of moving a shop, it is, perhaps, more necessary to create a spirit of cooperation among the men than it is at any other time, and to obtain this, many firms offer a bonus to the men for every day under a certain time that the shop is in running order, the bonus to be divided between the men in proportion to the work and responsibility assumed by each at the time of moving. This never fails to impart new energy to everybody, and even the office boy is imbued with new vigor in getting a shop cat into a bag to be taken to the new place. After the shop has been moved, the office quietly goes the same way as the shop. The main point to be considered in moving the office is to keep track of all records, as it often happens at the time of moving offices that old, and presumably useless, records are thrown away and lost.

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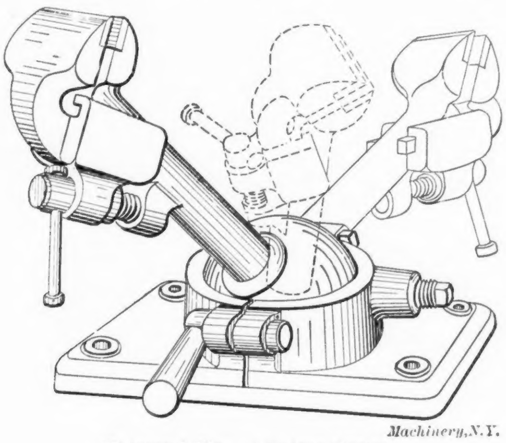
ITEMS OF MECHANICAL INTEREST.

UNIVERSAL VISE OF UNIQUE DESIGN.

At the Olympia Exposition in London, which took place during the latter part of last year, a vise of unique design was exhibited by Wadkin & Co., Leicester, England. The accompanying cut gives an idea of the manner in which this vise may be used, the general principle of the vise being plainly in

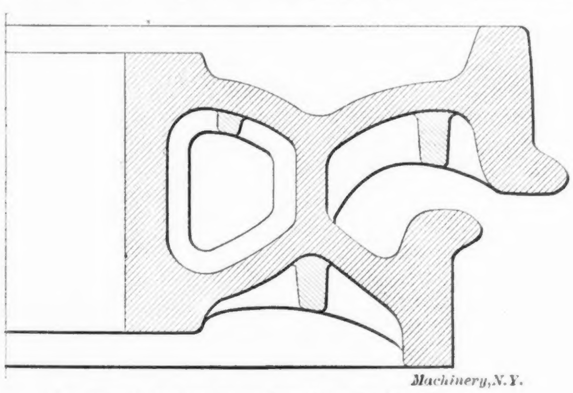


Universal Vise of Unique Design.

evidence. It can be swiveled around to almost any position, and tightened in place wherever desired. Considering its universal features the vise is said to be unusually rigid and effective for all ordinary bench work. The jaws are carried on the upper end of an arm, the lower end of which is fitted into a base having a ball and socket joint. It is of great advantage when working on irregularly-shaped pieces, as it is not necessary to change the grip of a piece in the vise or to work at the vise in an uncomfortable and awkward position, but the vise itself can be swiveled around to bring the piece clamped into the position required for convenient application of the tools used.

NEW IDEA IN CAR WHEELS.

The accompanying cut, taken from the *Railroad Gazette*, November 8, 1907, shows a new car wheel patented by Mr. P. H. Griffin, president of the New York Car Wheel Co., Buffalo, N. Y. The purpose of this wheel is to divide the destructive effects of heavy wheel loads and the wear and heating action of the brake shoe between two similar treads, and thus more than double the life of the wheel. The outer tread, as shown in the cut, is of large diameter, and rolls on the rails, carrying the load. The inner tread, of smaller diameter, is used only for braking purposes. This construc-



Car Wheel having Extra Tread for Braking Purposes.

tion requires a longer hub than the ordinary wheel and adds between 200 to 250 pounds to the weight of a 700-pound wheel, or approximately from 30 per cent to 35 per cent. The objections to this wheel, however, cannot be lightly dismissed. The weight of car wheels already adds considerable to the dead weight of a car or a train, and so considerable an addition to the weight as necessary in the car wheel described naturally is an important objection. There is also another point worth considering. The wear on the tread of a car wheel caused by its rolling on the rails soon forms a groove in the tread, and the action of the brake shoe has helped to eliminate the continual deepening of this groove by wearing evenly over the whole surface of the wheel. Dividing the action of the rail and the brake shoe will eliminate the beneficial action of the brake shoe in preventing a groove to be formed on the tread of the wheel, and it is likely that the tread rolling on the rails will have to be trued up a great deal oftener than would be the case in ordinary wheels.

DIFFERENTIAL DRIVE.

The accompanying cuts, taken from an article in *Der praktische Maschinen-Konstrukteur*, show the principle and application of a differential drive, or what might be called a geared counter-shaft. While the principle in itself is by no means new, the present construction is of some interest. If we first

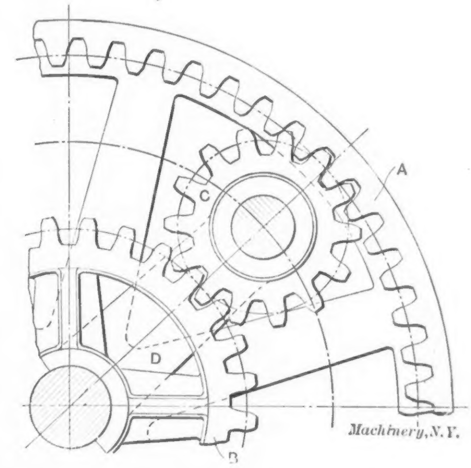


Fig. 1. Differential Drive of Interesting Construction.

refer to Fig. 1, it is clear that three cases for the use of this device may be conceived of. In the first place, the internal gear *A* may be stationary, and the gears *B* and *C* revolved; the gear *C*, of course, is simply rotated loosely on a stud fastened to an arm *D*, so that, in fact, it is the motion imparted by the arm *D* that is transferred to the gear *B*, or *vice versa*. In the second place, the arm *D* may be stationary, in which case either the internal gear *A* or the gear *B* may be the driver. Finally the gear *B* may be stationary, the power being transmitted in either direction between gear *A* and arm *D*. Fig. 2 shows a practical application of the principle. In

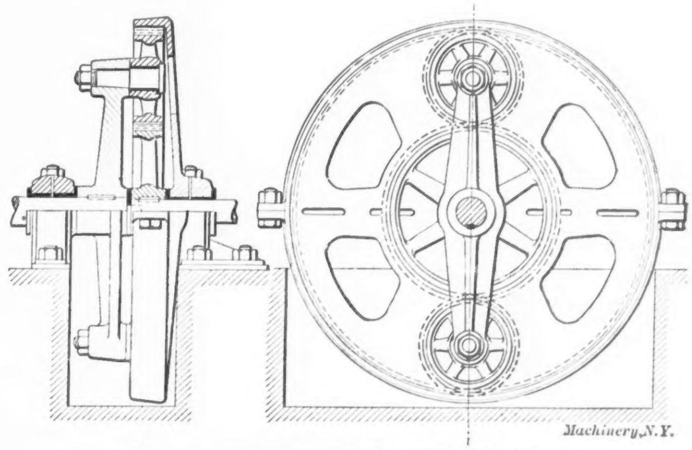


Fig. 2. A Practical Application of the Principle.

this design there are two gears corresponding to *C* in Fig. 1, mounted on an arm which in turn is keyed to the end of a shaft. The internal gear is stationary, forming part of a casing inclosing the gears, and the central gear is keyed to another shaft and drives this at a speed different from that of the driving shaft. This arrangement is a compact one, and is valuable, where space is limited, for the driving of a shaft directly in line with another shaft, but to be run at a different speed.

* * *

It is stated in *Copper and Brass* that aluminum bronzes containing less than four per cent of aluminum can be easily worked, but a greater amount produces a metal that is very hard to machine.

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THE LIGHTING OF DRAFTING ROOMS.

In no part of the office is the absence of direct shadows more to be desired than in the drafting room. High studding, with windows on the north side reaching to the ceiling, or a saw-tooth roof with the glazing facing the north, give the best and most uniform light. With the former, the arrangement of tables for a left-shoulder light is more vital than with the latter. Prismatic glass in the upper window sash is often of inestimable value in lighting distant corners, but it must be avoided when it would receive the direct rays of the sun. The tone color of the walls should be warm and restful to the eye, with the maximum reflective power. Plaster is usually much to be preferred to wood. The method of artificial lighting to be employed depends largely upon the character of the work. If large drawings are the rule, the light must be thoroughly distributed, as may be best accomplished by properly shaded overhead enclosed arc lamps. If the drawings are small, calling for little change in the area to be lighted, local illumination is very satisfactory, such for instance as by means of a single 32-candle-power lamp, swung from the back of the table upon a double-jointed bracket. This light, well shaded, and usually below the level of the eye, gives maximum intensity and lighting efficiency. The general appearance of the room is much better than with suspended lights in any form.

* * *

BORROWING TOOLS.

One of the little things in the shop which causes a great deal of lost time to the employer, and much annoyance to the more thrifty and careful of the men, is that of borrowing tools. Some men seem to think it perfectly proper to have their fellow workmen supply them with almost any tools they may happen to need, from a screw-driver to a micrometer. They have always some excuse to offer why they have not supplied themselves with a certain tool, and are always saying, "Guess I will get one myself," but they seem to forget it the next minute. When the rightful owner is in need of his tools, they are not on hand, as the habitual borrower seldom remembers or cares to promptly return what he has borrowed, and sometimes does not know what he did with it. The result is a constant searching for tools, with the consequent loss of much time. Sometimes the tools are entirely lost, and the borrower will assert that he has returned them. These matters may seem trifling as a subject for editorial comment, but we are convinced that if the time lost through this practice of borrowing tools could be estimated, and expressed in dollars and cents, the aggregate would be staggering in amount, particularly in large shops with a great deal of floating help.

It is true that the machinist has to supply himself with an expensive outfit of tools if he wishes to be well supplied—so expensive, in fact, as to be rather out of proportion if compared with the average income of machinists as a class; but that is no reason why one man should take advantage of his fellows. Tools cost one workman as much as another, and the borrower usually can afford to buy them just as well as the lender. Borrowed tools are not always handled gently, and it is an imposition to expect a man's friends to pay for the tools he uses. If each man tries to be independent of the other in this respect it will add materially to the efficiency of the entire force, and when it is really necessary to borrow, or help of any kind is needed, the infrequency of the necessity will induce a greater spirit of cooperation.

* * *

ON KEEPING THINGS SEVEN YEARS.

"Keep a thing seven years and you will find a use for it" is a current saying, for which almost any one can find warrant in his own experience. People who believe thoroughly in this principle have their desks and their houses filled with a collection of articles of more or less value which they are saving for the seven years period, with the reasonable expectation that a use will be found for some of it, at least, within that time. Such collections may be arranged in orderly fashion or heaped in wild confusion, depending on the characters of those responsible for them.

This same condition may sometimes be seen in a shop where the superintendent is imbued with this idea of preservation. In such a place there will be found, in more or less orderly arrangement, large collections of scrap, which from time to time furnish material for use in repair work, and from which old pulleys and gears are dug out to be worked over into special machines. Carrying out the same idea, old machinery, which has outlived its usefulness so far as continuous production is concerned, is kept in commission for the sake of the occasional odd job for which it is particularly suited.

Now, is not this idea a fallacy, however successful it sometimes may appear to work out in practice? It takes valuable time to search through a collection of such material for the one article needed, and the material itself requires valuable room if carefully and systematically laid away. If scattered indiscriminately over the shop, as is usually the case, it is a source of confusion, and interferes with the work, and such wreckage lying around loose cannot fail to have a demoralizing effect on both superintendent and workman.

The question to be considered is whether order and system, and the ability to do the work properly, are not of greater value in the long run than the money represented by an occasional useful article which has to be sacrificed for its price as scrap.

* * *

Probably reinforced concrete building construction requires just as careful designing and supervision as any other form. An engineer connected with one of the prominent contracting engineering companies of New York says that the apparent ease with which a reinforced concrete building can be erected has caused the business to be taken up by a large number of inexperienced contractors. Essentially, a reinforced concrete structure requires careful designing and engineering, and in addition, a most thorough and conscientious system of intelligent inspection, while in course of construction. Reinforced concrete, as well as any other form of construction, should be executed only by contractors experienced in that particular line of work.

UNIFORMITY IN SHOP FORMS.

Before the advent of the cost accountant and the betterment engineer, with their cost index and numerous blanks, most shops displayed a heterogeneous collection of shop forms, devised by different individuals, suited to the needs of different departments, and printed at different times by different printers. The trend is now decidedly toward uniformity in material, size, and type. Variety in color of ink and paper is by no means as necessary as sometimes appears. Judicious selection, and reduction to the least number of varieties, will permit of greater economy in printing, inasmuch as a number of different shop forms may be run together on the same stock, or in the same color of ink. The same is true of the sizes, which may be made in multiples, so that a similar grouping will reduce cost in printing. There is always a strong tendency to make a system absolutely complete by providing a printed form for each type or class of report. This results in great and often unnecessary multiplicity. Particularly in the case of index cards always retained in a given department, all printing may be omitted when the meaning of each item or notation is perfectly clear to those who habitually refer to the cards. It is often the case that expensive ruled sheets are required for monthly or annual reports. The number is so small that carefully hand-ruled sheets, or blue-prints will serve fully as well, with a material saving in the cost as compared with type composition, printing, and machine ruling. A careful consideration of the conditions under which they are used, and a realization of the fact that they are not made for strangers to the system, will frequently result in a decrease in the number of printed forms, and a material economy in the entire expense of shop stationery.

* * *

THE DECEMBER MEETING A. S. M. E. PAPERS.

Contrary to our usual custom, we have not made a general abstract of the papers read before the December semi-annual meeting of the American Society of Mechanical Engineers. The number of papers presented would have necessitated brief paragraphs of a somewhat perfunctory nature had the abstracts been confined to ordinary limits of space, and the general character of the papers makes them of minor interest to the majority of our readers. No disparagement is intended, nor is it merited. The papers on the whole were of a high grade, and they constitute a notable lot of contributions to engineering literature; but we cannot help feeling that they possessed too little variety. Out of a total of nineteen papers presented at the December meeting, all but two were either on power production and closely kindred subjects or foundry practice. The December proceedings are rich in material for the power plant engineer and the foundryman, but rather dry picking for the machine shop superintendent or machine designer.

Undoubtedly the most effective results for a favored group can be obtained by these symposiums on given subjects, but it should be remembered by the committee on papers that the membership is made up of men representing a great variety of engineering interests. The production of power is important; in fact, it is a basic art on which all other industries depend. The manufacture of castings in a sense is also a basic art and the foundation of nearly all machine work, but there is a vast multitude of other things of interest to the mechanical engineer which also possess great importance. We should like to see more variety in the papers presented, and more of the lowly things that we all can understand and talk about. Power plants, superheated steam, gas engine development, and foundry practice are subjects of perennial interest, no doubt, but let us also have papers on machine work, machine shop practice, the drafting-room, notable examples of machine production and even shop kinks. In the same way, other fields of mechanical engineering activity can be treated so as to give the papers a diversity of interest that will make every member of the A. S. M. E. feel that his membership is profitable to him, and that it has a direct vital connection with his business.

BALANCED INDUSTRIAL CONDITIONS.

In time of peace, prepare for war! This maxim has been repeated so many times that it is now nearly universally accepted, although the feeling is growing that if we did not prepare for wars we should be less likely to have them. Whether this maxim be true or not in regard to peace and war, it is not open to denial that a similar expression can be applied to industrial conditions, and dealing with times of prosperity and of depression. As thus applied, the maxim would naturally read: In time of prosperity, prepare for depression. This, however, is a pessimistic view of industrial conditions, and the value of the suggestion is somewhat negative. We would prefer to phrase the maxim: In time of depression, prepare for prosperity. This is the optimistic view—a positive expression looking forward to increased possibilities and development, and expressing a different attitude towards the establishment of balanced industrial conditions.

Our industries, and in particular the machinery industry, have developed so rapidly that truly balanced conditions have not yet been fully accomplished. But it is no exaggeration to say that an increased confidence in the future, permitting, in times of less active business, necessary preparations to be made for the returning tide of business development and prosperity—which we all know to be ahead of us as soon as the temporary ebb has passed by—would greatly tend to balance industrial conditions. If improvements were carried out in dull times instead of being curtailed, not only could they be installed for a smaller outlay, but they would be available when active business required. We have all seen cases where our shops were unable to take proper care of their orders in good years, and time and money were wasted trying to turn out work with inadequate facilities; and while there is a natural reluctance to invest more capital in a plant when business is dull, foresight and provident business management often demand it in order to take advantage of prosperous times when they return. In every sound business the approximate relation between present expense and future requirement can easily be judged, because there can be no doubt of the ultimate return of prosperous conditions.

One of the great differences between European and American industrial management is to be found in the conditions under consideration; and it may be that the balance generally maintained on the Continent depends somewhat upon the manner in which most European shops add extensions. Unlike many of our own manufacturers—who are often forced, we admit, by the unprecedented demand for their product—European manufacturers usually add improvements when times are dull and when labor and materials, as well as interest, are low, and the demand on their facilities is light. In good times they devote themselves exclusively to the manufacturing business, being then equipped to take care of all the trade they obtain, and naturally making a larger profit on their product. The additional expense of carrying a building or two, not yet equipped to its full capacity, for a couple of years, they claim is amply justified by the certainty of increased returns later. There is, perhaps, a lesson in this for us, which could be applied to our present conditions. Instead of increasing our plants when times are prosperous and everything expensive, and completing the installation of the equipment about the time when trade begins to fall off, would it not be possible, to some extent at least, to prepare for increased trade before the demand exists? The question has but one answer. If we look far enough ahead, we can see the tide turning, and now is the time to prepare for it, rather than to lose heart on account of present conditions.

* * *

In a report recently presented to the General Managers' Association of Chicago, it is stated that on American railroads the number of locomotives has, in the last ten years, increased by 42.8 per cent. In the same period the weight of some types of locomotives has increased 82.8 per cent, and the average weight increased 28 per cent. The number of freight cars has been increased by 51.5 per cent in this last decade, their total capacity 95.3 per cent, and average capacity 30.6 per cent.

ENGINEERING REVIEW.

CURRENT MECHANICAL EVENTS—LEADING ARTICLES OF THE TECHNICAL PRESS.

According to *Industritidningen Norden*, an international exposition will be held in Turin, Italy, in 1911. This exposition will be large in its scope, and will be devoted particularly to industrial products and agriculture.

It is reported from Washington that the Postmaster-General has imposed on a number of railroads the penalty stipulated in their contracts for failure to deliver the mails on time. The Great Northern Railroad has been fined more than $26,000 for slow deliveries from points in Minnesota to points in the state of Washington.

The inventor of the mono-rail gyroscopic railway car, Mr. Louis Brennan, has been granted a sum of $30,000 by the Indian government, in order to enable him to proceed with his experiments with a full-sized mono-rail car which he is now constructing for that government. The principle of Brennan's mono-rail car was described in the August issue of MACHINERY, engineering edition.

A company has been organized, and a factory for the manufacture of glass telegraph poles has been built at Grossalmerode, a town near Cassel, Germany. The glass mass of which the poles are made is strengthened by a reinforcement of steel wires. One of the principal advantages of these poles is their use in tropical countries, where wooden poles are soon destroyed by insects and where climatical influences are ruinous to wood.

Contracts were closed some months ago for the construction of floors, interior partitions, and roof of the Pennsylvania Railroad's new passenger terminal in New York City. The material used will be absolutely fireproof, and porous terra cotta has been selected as the only material that could stand the required tests. This material will be used in form of hollow blocks as a complete covering for the steel frame of the building.

A Canadian correspondent to the *Times Engineering Supplement* states that on the new National Transcontinental Railway, known also as the Grand Trunk Pacific, a section of the line, 70 miles long, will be constructed without a curve, and also without a rock cut, no obstruction being met with except rivers to be bridged. The longest tangent on any railway in the world is to be found in Argentina, where, on one road, there is in one place 175 miles of straight track.

Platinum has recently been discovered in California, and according to the *Railway Age*, the United States Geological Survey is at the present time investigating these discoveries. The investigations have gone on for a year or more, and it has developed that there is platinum in 120 places in the United States. The increased commercial use for platinum and the high price of the metal, owing to its scarcity, make these investigations very valuable. Should the natural deposits of platinum in the United States be found to be of any considerable amount, they would constitute an addition to the known natural resources of the country of no small importance.

A contributor to the *Engineering News* tells of a neat method he has followed for lettering maps and drawings which are to be reproduced photographically by the zinc process. Instead of spending the time to do the lettering carefully by hand, he has the various names, titles, etc., set in suitable type and has proofs taken on gummed paper. Strips containing the words and titles are cut from this gummed paper and pasted on the drawings. These labels do not appear in the finished cut, nothing being reproduced except the printing. This gives neat, uniform lettering at a less expense than would be required for work of a similar quality done by hand.

It is stated in the *Engineering Record* that the preservation of ties by crude oil has been successfully tried by the Atchison, Topeka & Santa Fe Railway. The company some years ago treated a few ties with California crude oil containing about 75 per cent asphaltum base, the remainder being light oils, the greater part of which vaporized when heated. The oil was heated to 180 degrees F., and forced into the wood under 150 pounds pressure per square inch, the ties taking up from 4 to 8 gallons each. They were placed in the track at the section where untreated ties will not last over two or three years, but the treated ties, when examined after having been in service for 4 years and 9 months, were found perfectly sound.

The increasing application of internal combustion engines, fed with coal gas, blast furnace gas, and other gases, has, says *Engineering*, unfortunately, been accompanied by a large number of cases of gas-poisoning. In Germany both the Imperial Insurance Office and the Board of Health have drawn the attention of manufacturers and engineers to this source of danger. The attendants of gas engines should be cautioned, and be instructed as to the measures to be adopted in accidents. The most dangerous constituent of the gases is the carbon monoxide, and cures can be effected by making the victim inhale oxygen. Oxygen cylinders provided with suitable valves should be kept where gas engines are used, and some attendants be duly trained by medical men in applying the first aid.

A feature of special interest at the Olympia exhibition in London was that of a line of compressed paper pinions, manufactured by the British Insulated & Helsby Cables Co. These pinions are intended to compete with the raw hide pinions, and it would seem from records of the samples shown that they are likely to come into great popularity. The specimens exhibited were pinions which had been in use under hard load conditions and unfavorable surroundings for periods long enough to prove their ability to stand up in hard service. The makers claim that these paper pinions are unaffected by moisture and oils. The makers have had a successful experience with compressed manila paper for insulation of cable work, and, judging from this experience, it is reasonable to expect that the paper pinion has been evolved on thoroughly firm ground, and with reasonable assurance of successful working.

In an article in the *Zeitschrift für Angewandte Chemie* a new method for manufacturing copper tubes is described. This method is in operation at the works of Langbein & Co., in Leipzig, Germany. The copper is deposited on a revolving mandrel as in the Cowper Cole's process, but the high rate of revolution of this latter process is dispensed with. The density and smoothness required in the deposit are obtained by suspending in the solution some siliceous material such as sand or quartz, which is unacted upon by the electrolyte, and which supplies the necessary degree of friction to the surface of the mandrel to remove all gas bubbles and inequalities in the deposit. Tests of the tubes made by this process have shown that the tensile strength is twice as great as that demanded by the specification for copper tubes drawn up by the authorities of the German Marine. The method can be applied to the production of plates and tubes of other metals than copper. In fact all metals that are liable to electrolytic deposition in a spongy and porous state can be obtained in a dense and coherent deposit by use of this method.

It is stated in the *Brass World* that the most frequent cause of the cracking of aluminum castings is the overheating or "burning" of the aluminum while melted. The most generally used aluminum alloy for castings, at the present time, is one which contains aluminum and zinc, and occasionally copper. The presence of zinc renders the casting more liable to crack,

but, on the other hand, the greatest strength in the metal is obtained by using zinc. If the following rules are adhered to, less trouble will be met with in aluminum castings. In the first place, the metal should be melted with a slow fire, so that the top of the metal will not be burned before the remainder of the metal is melted. When the whose mass is melted, care should be taken not to overheat. The ingots should be packed in the crucible as compactly as possible, so that portions of the metal will not stick up and become exposed to the action of the flame, and the most important rule may be said to be not to have the aluminum melted before the mold is ready. Non-observance of this last requirement is perhaps the most common source of burned metal, and consequently the most frequent cause of the cracking of aluminum castings.

The English government, according to the *Times Engineering Supplement*, is at the present time making preparations for introducing oil for fuel for steam raising purposes in the navy, on a large scale. Oil storages have been provided for in several localities. Among liquid fuels considered, is one known as "masut," which is one of the by-products of the distillation of raw petroleum, and which is now used to some extent by the Russian, German, and Italian navies. As it appears to have a high calorific value, and is said to produce but little smoke, it has some of the chief requisites for a fuel to take the place of coal. The relative cost of production of masut and of coal would probably be disadvantageous to the former, but this would be offset by the decreased cost of transportation and the conveniences attending the use of fuel in a liquid form. It is likely that the future will pay more attention to liquid fuels than has the present. The natural supply of liquid fuels is abundant, it being stated that Australia has immense deposits of oils which can be used for fuel purposes, which have as yet not been exploited at all.

In an editorial in the November issue of MACHINERY we found occasion to touch upon the future of the automobile, and we also mentioned that we expected that before long the more or less useless pleasure vehicle would be followed by vehicles for actual commercial use. It is gratifying to find that this sentiment is now voiced by automobile manufacturers themselves. In a recent issue of the *Horseless Age*, the manager of a well-known concern is quoted as having said: "I feel more and more strongly drawn to the commercial end of the business, and am convinced that in time we shall make trucks exclusively. Nevertheless, I do not think it advisable to give up the sale of pleasure cars at the present time." It is also stated that this awakening to the possibilities of the commercial vehicle is general, and that there is scarcely a company of any prominence connected with the industry which has not begun to take a more or less active interest in motor cars for trucking and general use. Some have designs under way, and others have various experimental vehicles on the road.

It is well known to anybody who has attempted to make a good photograph from a machine having bright polished surfaces as well as dark painted portions, that the bright portions cause trouble, and there is difficulty in showing these portions in a proper manner, the work of exhibiting the machine clearly being largely involved upon the retoucher. If the exposure is made of such duration that the bright portions will be clear, there will be under exposure of the dark painted portions, and if the exposure is made to suit the darker portions, then the bright portions are usually not plain, and sometimes entirely indistinguishable. In *Industritidningen Norden* we find mentioned a method for so preparing the bright portions of a machine to be photographed that a satisfactory picture can be taken, which will show all details with an equal degree of exactitude. This method consists in subjecting the bright surfaces to vapors from sal-ammoniac, which gives to the surface a fine covering which takes away the brightness of the surface. If the sal-ammoniac is pure (neutral) the vapors have no effect whatever on the machine, or on the surfaces themselves, and the covering can easily be wiped off with a cloth without using either oil or other means for removal.

FORMULAS FOR GAS ENGINE CRANK-SHAFTS.

During the last few years, although manufacturers of gas motors have materially increased the diameters of the crank-shafts of their engines, these have still proven themselves too weak in many cases for the stresses imposed on them, owing to the higher compressions and heavier fly-wheels in present-day designs. While a few years ago, in gas motors, an explosion pressure of 250 pounds per square inch was seldom exceeded, in present practice 400 pounds per square inch is not uncommon. The stresses on this class of machinery are always more or less uncertain, and therefore calculations of strength are difficult, and dimensions have largely been determined by experiment, or, in other words, by trial and error. For this reason some remarks by Mr. Michael Longridge, as published in the *Mechanical Engineer*, are worth noting. According to his opinions, while reliable theoretical formulas are not deducible, the sizes which experience has shown to be desirable can be expressed fairly accurately empirically. If C is the diameter of the cylinder, D and d the diameters of the shaft and crank-pin journals, respectively, r the radius of the crank, and L the distance between the inner edges of the shaft journals, that is, the distance over-all of the crank webs, all expressed in inches, then for engines below 12 inches diameter of cylinder the value of the expression

$$\frac{C^2\,(L + d)}{D^3}$$

should not exceed 12 for engines with two bearings and overhung fly-wheels, or 14, if three bearings are used. Furthermore the value of the expression

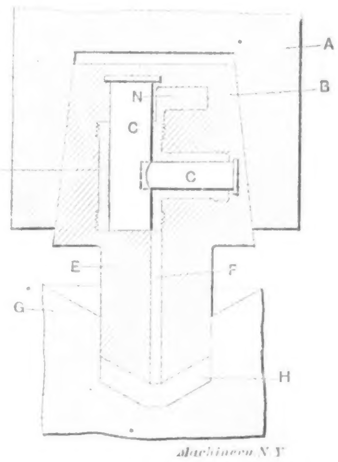
$$\frac{C^2\,r}{d^3}$$

should never exceed 10. Besides, the crank webs should always be joined to the fly-wheel shaft with large fillets, as sharp corners have proven to be a common and serious source of weakness.

CASTING METALS UNDER PRESSURE.

The Mechanical Engineer, November 16, 1907.

By the process of producing castings under pressure, it is possible to make the castings of complicated shapes and with such details produced by the casting process as it has not heretofore been possible to form excepting by finishing with special tool. The processes of casting metals under pressure are divided in two groups. In the first of these groups the pressure exerted upon the material being cast is produced by means of gas, while in the other case the pressure is exerted by mechanical means. The device illustrated in a schemetrical form in the accompanying cut shows the appliances used in a process patented by Mr. F. Ljungström, Flemminggatan 8, Stockholm, Sweden.



Arrangement for casting Metals under Pressure

Referring to the cut, A is a piston operated by hydraulic pressure, at the lower end of which the mold B is inserted. The mold is intended for a T-pipe, C being the core of the pipe. The mold is provided at its lower end with a piston-like extension E provided with a taper at its lower end, and also with an internal passage F. This extension fits into the hole H in the part G. The hole or cavity H is filled with molten metal, and the piston A with the mold B is moved downward so that E enters H, and thus exerts a pressure upon the molten metal. The metal will then rise into the cavity D, and the desired casting will be formed under pressure. As the mouth of the passage F is located at the lowest level of the end of the piston-like extension E, the layer of oxide collecting on the surface of the molten metal in H will not enter the

passage. If, however, a few particles should penetrate into the passage they will collect in the pocket N, branching from the actual form of the piece to be cast. The advantage which is claimed for this method is that the mold does not need to be heated before the molten metal is introduced, and for this reason the frame of the mold can be made out of metal, so as to give the required strength required by high pressures. In previous designs of this kind it has been necessary to heat the molds, before introducing the molten metal, to a temperature as high as the molten metal itself, and in cases of metals at a high fusion temperature, it has not been possible to make the molds from metal or alloys, but they have been made from plaster of paris, asbestos and similar materials which could not stand high pressure.

A YEAR'S EXPERIENCE WITH GAS ENGINES.

Abstract of paper by Mr. Paul Winsor, read before the American Street and Interurban Railway Engineering Association at Atlantic City, N. J., October 16, 1907.

The author of this paper is chief engineer of motive power and rolling stock of the Boston Elevated Railway Co., and is therefore well acquainted with the subject in hand. The gas engines in question were installed in the plant of this company early last year at the Somerville power station, the equipment consisting of two 600 B.H.P. 2-cylinder 4-cycle gas engines with generators and gas producers. The plant was started in May, 1906, and since then has given continuous, reliable, and satisfactory service, the engines being run sixteen hours per day. The fuel used for gas generation has been soft coal, the coal used per kilowatt hour averaging 2.034 pounds or 1.404 pound per B.H.P. hour. Some trouble was experienced with ignition. The igniters had originally platinum tips, and these were rather expensive to replace. For this reason the plant has been run for four months without any platinum tips, and less trouble has been experienced. During the first months, pre-ignition and back fires were rather frequent, but by lowering the compression in one of the cylinders, and changing the igniters, these troubles have been greatly reduced. The reliability of the service has proved one of its strong points. The plant can be started up at any time in less than five minutes. The gas engine station used only 46.1 per cent of the coal used at one of the steam plants of the same company per brake horse-power. For this reason the author concludes that a gas engine plant equipped with gas producers will operate fully as reliably as a steam plant, and will require from 30 to 60 per cent less fuel. The drawbacks to be considered would be the first cost which is about $200 per kilowatt when rated so as to have 33 1/3 per cent overload capacity, and also the present small size of gas engine units.

THE CRYSTALLIZATION OF STEEL.

Walter Rosenhain, in the *Times Engineering Supplement*, November 6, 1907.

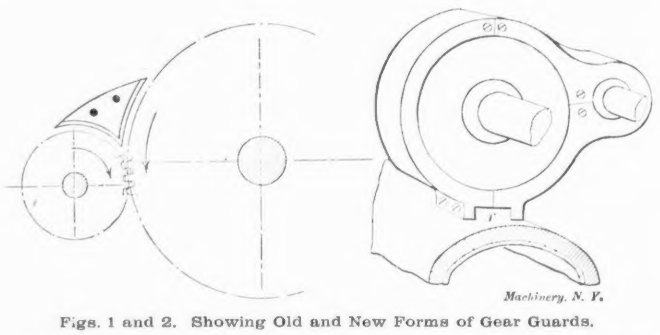
In our issue of MACHINERY, engineering edition, December, 1906, we published a short statement regarding the crystallization of steel, in which it was contended that steel does not crystallize by use, but that in cases where steel, having given away under stress, shows a crystalline structure, this crystalline condition has been inherent in the steel, and the use of the steel simply separates the faces of the crystals. In the present article the author sustains the same theory. The author states, in the first place, that microscopic study has proved beyond doubt that all metals possess, in any state, a truly crystalline structure, and that, therefore, ordinary materials of construction, particularly iron and steel, cannot be said to possess the fibrous structure, as so often has been contended. One cause of misconception regarding the character of metals lies, no doubt, in the fact that we are accustomed to regard crystal as brittle bodies, easily broken or split. This brittleness, however, is only the property of one class of crystals. It has been shown, for example, that under suitable conditions such a perfectly crystalline substance as marble can be caused to flow under stress and undergo drastic deformation without fracture. Ice is also a typical crystalline body which shows a great amount of plasticity. The fact that metals are characteristically crystalline is, however, not in any way in conflict with our knowledge of their ductility and strength. When fibrous fractures appear, it is simply because the crystals have been subjected to deformation, before the fracture occurs, by the application of the stresses in a certain manner, while on the other hand, when sudden and intensely local forces develop, the crystals will split or cleave along their natural cleavage before any amount of deformation has taken place. In such cases, we have the fractures which are claimed to have occurred on account of the crystallization of the metal, although the crystalline or fibrous character of the fracture is dependent not as much upon the character of the metal itself as upon the manner in which the fracture has been produced.

The author finally concludes that inasmuch as metals, in their normal condition, already have a truly crystalline structure, it is evident that exposure to vibration cannot produce such a structure, although it might modify the character of the crystals already existing; but there is, as far as the author knows, at the present time, no evidence at all that any change in the size or arrangement of the crystals of a metal can be produced either by vibration or fatigue. The decisive and final abandonment of the "crystallization" theory thus is evidently unavoidable, and some of our cherished misconceptions will have to be abandoned. At the same time, it must of course, be admitted that fatigue fractures reveal more clearly the crystalline structure existing in the metal, and it is also evident that fractures would appear at sudden stresses more easily in metals of coarse crystalline structure. This is in itself a point of considerable practical importance, and it explains to a great extent the prevalence of the crystallization theory, for it gives us the reason why faulty conclusions have been arrived at.

GEAR GUARDS.

Zeitschrift des Vereines deutscher Ingenieure, August 10, 1907.

In an article about the development of German machine tool design, the author, Mr. Ruppert, devotes himself also to the influence on the design of machine tools of the German laws for protecting employes from injury. He calls attention to the fact that these laws, while at first not considered to be of any consequence to the machine tool builder in any other respect than that he had to safeguard dangerous gears and belts in his own shop by a piece of stiff sheet-iron or wooden board, finally have proved to give a distinct characteristic to machine tools on which every gear and projecting revolving part is covered with a suitable guard. The law covering compensation to injured employes is far more detailed than that of the United States, and it probably is for that reason

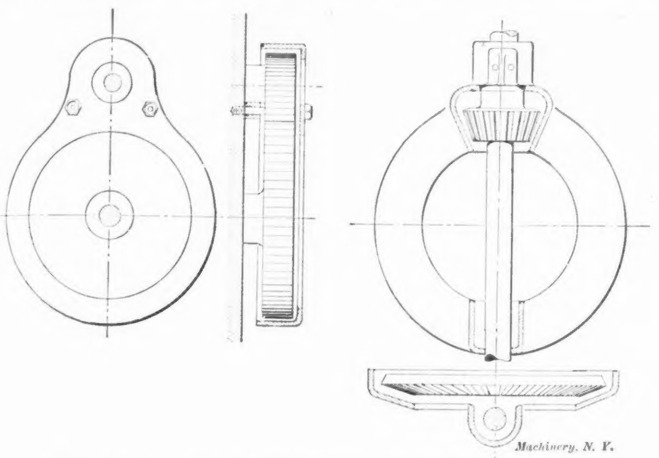


Figs. 1 and 2. Showing Old and New Forms of Gear Guards.

that foreign machine tools until lately always have been far better protected than American tools as a rule have been. The German law stipulates that the owner of any machine is responsible for any injury inflicted upon any person in his employ working with his machine. While at first, as mentioned, it was not expected that this law would influence machine tool building to as great an extent as it has, it was soon realized that machine tools had to be provided with suitable safeguards in order to sell as well as other machines well safeguarded, but not superior in other respects.

The first crude form of gear covering as used in Germany is shown in Fig. 1, and was not unfamiliar to American products a few years ago. It consists simply of a small piece of bent sheet iron, or at best, a thin casting covering the portion where two gears engage with one another. This inefficient

guard was probably adopted for economical reasons, it being considered too expensive to cover up the whole gearing. If the gears, however, are running in a direction as indicated by the arrows, this guard is really almost as dangerous as the unprotected gearing, as the fingers may be pulled in



Figs. 3 and 4. Guards which cover only the Toothed Surfaces of the Gears.

between the gear and the guard almost as easily, or perhaps more easily, than they would be liable to be drawn in between the gears themselves.

A few neat designs of a more recent date, however, are shown. Some of these are not merely covers but brackets which serve as support and bearings for the gear shafts, as well as for safeguarding covers. In Fig. 2 is shown a cover which completely encloses the gears, and in order to facilitate assembling, it has been necessary, in this case, to make the gear case in several parts. The design is a remarkable example of a gear guard made up from several parts, and still giving the impression of a uniform and neat design. In Fig. 3 is shown a guard of fairly similar design. This guard encloses completely the outside of the toothed surface of the gears, while it is open on the inside and in the central portions on the outside so as to permit of easy assembling of the machine. Fig. 4 shows a cover for bevel gears in which the cover also completely encloses the toothed portions of the gears. In Fig. 5 is shown the neatest and most modern design for covering miter gears. In this, the guard itself and the bearings for the shaft carrying the gears are cast in one integral piece, the gears being assembled from the outside of the guard, which is left open. After being assembled, a cover, as shown by the dotted lines, is placed over the opening, thus completely enclosing the gears. These forms of guards for gearing are not new, but they have a particular interest as they show the development of the idea of the necessity of making the risks of industrial workers as small as possible, and some of them may contain an idea that can be employed in other cases similar to those shown as well. The designer of to-day is required to keep in mind the proper guarding of revolving machine parts, just as much as he is required to keep in mind the proper ratio of the gears themselves, and the cheapest and best guards for gearing are those which can be made up in a uniform style with the machine itself, and be designed so as to form integral parts with the machine. Wherever a guard becomes such a part of the machine it will serve its purpose to the fullest extent, inasmuch as there is then no risk of its being left off because of being inconvenient to the man operating the machine.

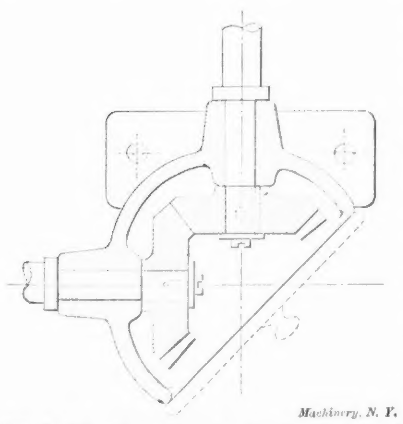


Fig. 5. Miter Gear Guard, with the Guard and the Bearing for the Shaft cast in One Piece.

NOTES ON ROPE DRIVES.

Thos. Hart in the *Mechanical World*, October 11, 1907.

At the beginning of this article, the author lays down the rule that one should try to avoid arranging rope drives at right angles or around corners, whenever it is possible to arrange the drive so that it may run straight. A rope drive transmitting power around the corner reduces the power probably by 50 per cent, while its life is shortened by even a greater amount. From this it is evident that this kind of drive is not economical, but there are cases where such drives are imperative, and the author has attempted to show the best methods of arranging these drives.

Fig. 1. Half-cross Drive.

Fig. 2. Rope Drive with Shafts at an Angle of Ten Degrees.

Half-cross Drive.

Fig. 1 shows what is called the half-cross drive. In arranging such a drive, the fundamental rule is that all the angle should be given to the tight or driving side of the rope, and the slack side should run straight. If the slack side took the angle, the sagging would cause it to leave the groove at the point of entry. When installing such a drive it is often good practice to fix one pulley and to let the other one loose on the shaft for a few minutes. It will then find for itself the position in which it will work the best. It is difficult to arrange this class of drive for more than one rope, and therefore this type of drive is seldom used excepting for small powers.

Drives between Shafts which are not Parallel.

Drives where the shafts are not parallel with each other are frequently met with, but there has as yet not been any positive answer to the question of the greatest angle at which the ropes can be made to keep in the grooves and do effective work. It has been stated that where shafts form an angle with one another of three degrees or less, a rope drive may be arranged which will run well without any special arrangement, but for larger angles special deviations from the standard shape of groove must be made. In Fig. 2 is shown a rope drive where the two shafts form an angle of 10 degrees

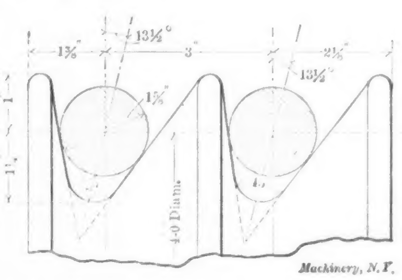


Fig. 3. Shape of Grooves for Rope Drive in Fig. 2.

with one another. It will be noticed that at the point where the rope commences to leave the groove, it requires more freedom than it can obtain in a groove of the ordinary shape, and for this reason the grooves have to be shaped in a special way. Fig. 3 shows the shape of groove adopted for the rope drive shown in Fig. 2, which is one that has been running satisfactorily for two years. In general, it may be said that rope drives of this character will give greater satisfaction if it is possible to have the pulleys of approximately the same diameter; if the distance between the pulleys is not too long, so that the sag of the slack side of the ropes becomes too great; and, finally, if the speed is not too high.

Drives at Right Angles.

The author understands drives at right angles to be drives around corners. In these, more than in any of the types

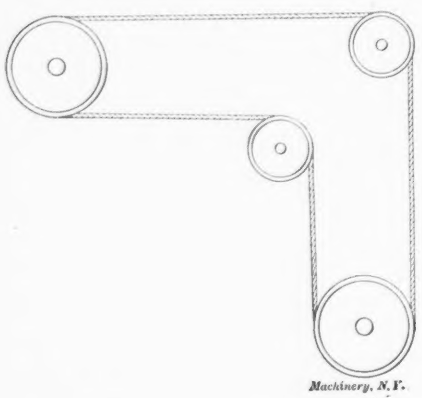


Fig. 4. Right Angle Drive, with Grooves Parallel.

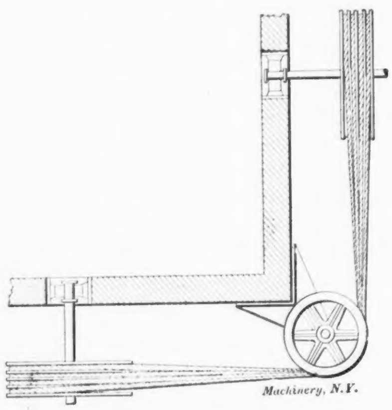


Fig. 5. Right Angle Drive, with Grooves at 90 Degrees.

previously discussed, is it necessary to assume that a great deal of power is lost through the bending of the rope in its passage from the driving to the driven pulley. In this class of drive, the driving and driven pulleys may have their grooves parallel to each other, although the ropes pass around the corner as shown in Fig. 4. This kind of drive offers no difficulties as far as the design or arrangement is concerned. A different problem is encountered in the case shown in Figs. 5 and 6. It is easily seen that the idlers here require different setting, and that careful thought is necessary to place them in the correct position. In Fig. 5 only the actual center of the driving and driven pulleys can be set true with the idler. Therefore, the greater the number of ropes, the greater is the tendency of the ropes to leave the grooves. Drives of three ropes seem to run satisfactorily with this arrangement, but when five ropes are used the first and fifth often cause

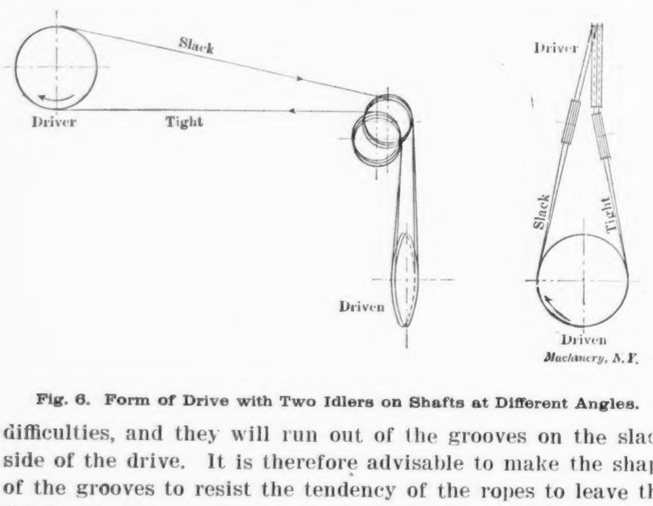


Fig. 6. Form of Drive with Two Idlers on Shafts at Different Angles.

difficulties, and they will run out of the grooves on the slack side of the drive. It is therefore advisable to make the shape of the grooves to resist the tendency of the ropes to leave the driving or driven pulleys in such cases. If one does not want to make the grooves with an inclined center line, it is well to make them very deep, as shown in Fig. 7. In Fig. 6 is shown a drive where the idlers are placed on two separate shafts in order to permit a proper position to be obtained for each of them to carry the ropes between the driving and the driven pulley. The grooves in the idler pulleys should be wide and round at the bottom, and the sides should be shaped so as to accommodate the rope at the position which it will naturally take in the groove. In the earlier days of rope drives, experiments were made with idler pulleys having no grooves, but having a flat bottom and outside flanges made to any angle suitable for the drive, as shown in Fig. 8. At first sight it appears as if this idea would be favorable to the types of drive under discussion, but on account of the sag of the ropes, particularly on the slack side, it often leads to one rope fouling the others, and possibly throwing them off the pulley altogether. The idlers should be made as large in diameter as possible, and should run in well lubricated bearings.

Loss of Power with Angular Drives.

The author concludes his notes on rope drives with repeating his reference to the loss of power in angular drives, and in that connection he refers to an actual case occurring in his practice. In this case, the drive was one of two ropes, each 1½ inch in diameter, in a spinning mill, arranged to drive around the corner on account of an objection to using bevel gears at this point. The trouble was that the ropes stretched considerably, and after tightening, quickly wore out, all this indicating overload. An inquiry showed that the two ropes were set to transmit about 60 horsepower. This load would have been easily carried in an ordinary straight drive, the speed being 4,500 feet per minute, but, on account of the angular drive, the ropes absolutely failed to carry this load in the case referred to. In another case five ropes of 1¼ inch diameter on a straight drive performed a certain work, but, owing to some alterations, it became necessary to alter the position of the engine, and to install a right angle drive. This was installed in the most practical manner possible, but it soon became necessary, in order to transmit the required power, to increase the ropes to a diameter of 1¾ inch, and still the life of the ropes was very much reduced.

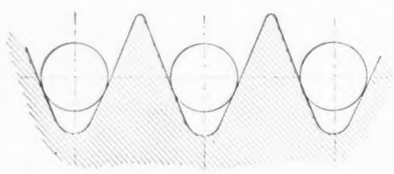


Fig. 7. Deep Grooves for Angular Drives.

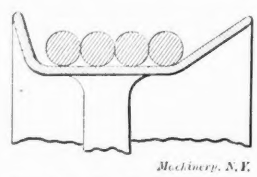


Fig. 8. Idler Pulley without Grooves.

For this reason it cannot be too strongly pointed out that drives such as those discussed should only be used under circumstances of great necessity.

BALL BEARINGS.—2.

Abstract from paper by Mr. Henry Hess, read before the American Society of Mechanical Engineers, May meeting, 1907.

Correct Ball Bearing Mounting.

The following requirements are based on correctly mounted ball bearings:

a. The proper size must be selected for the load and conditions in question. Rated capacities are usually for steady loads and speeds, but variations from these conditions demand a cutting down of the listed capacity.

b. Bearings must be lubricated. The often repeated statement that ball bearings can run without a lubricant is not correct.

c. Bearings must be kept free from grit, moisture, and acid. No lubricants developing free acids should be permitted.

d. The inner race of the bearing should be firmly secured to the shaft. This can be done by a light driving fit, reinforced by binding the race between a substantial shoulder and a nut.

e. The outer race must have a sliding fit in its seat. These conditions should under all circumstances be adhered to, and a failure to do so will result in very unsatisfactory bearings.

The two following conditions are frequently disregarded, and while the disregard of these conditions is not so serious as of those mentioned before, it is safe engineering to follow them, and a disregard of them is a standing invitation to trouble.

f. Thrusts should always be taken up whether in the same or opposite directions by the same bearing.

g. Bearings should never be dismembered, or at least never more than one at a time. That will avoid the danger of mixing the balls from different bearings; such balls from different bearings are very apt to vary more than is permissible for the individual bearing.

Illustrations of Correct Mounting.

The development of ball bearings being so recent is probably the cause that so little information as to correct mount-

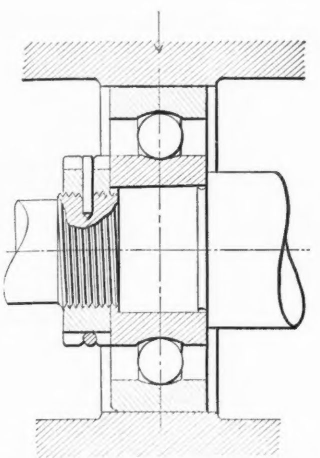


Fig. 16. Free Mounting for Radially Loaded Bearing.

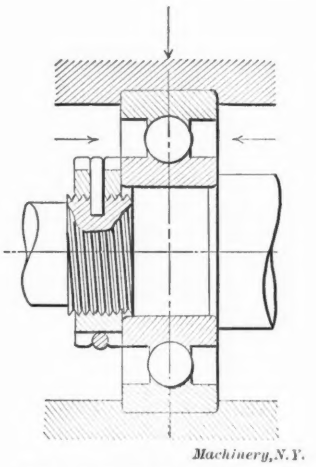


Fig. 17. Radially Loaded Bearing held against Longitudinal Motion.

ings for various conditions is available. The experience of the author of this paper has been that faulty mountings are very general, and for this reason he has given illustrations of correct mountings, going more into details than would be considered necessary for a more familiar mechanical subject.

Fig. 16 shows a bearing in which the inner race has a light driving fit on the shaft, and is securely clamped between a shoulder on the shaft and a nut. The shoulder on the shaft should be high enough to get a firm grip on the surface of the side of the race. It is good practice to make this shoulder about one-half as high as the race thickness, perhaps a little less for large bearings and a little more for small bearings. The outer race has a tight sliding fit in the housing, so that the bearing as a whole may be able to respond to relative shifting of the shaft and housing without being subjected to end thrust through the balls.

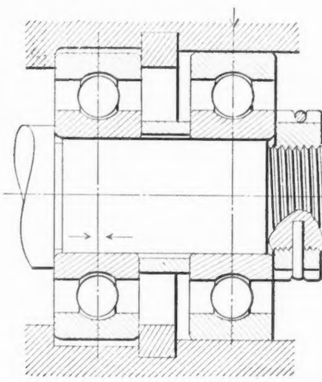
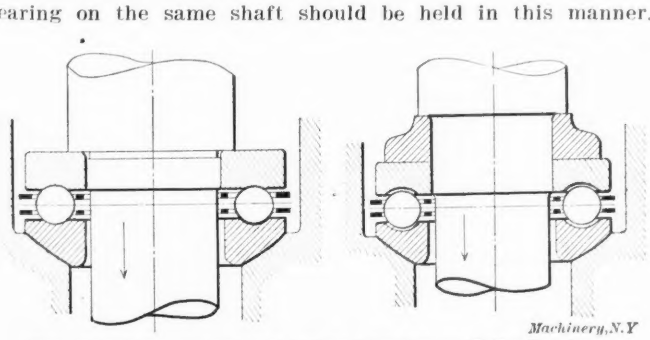


Fig. 18. Separate Radial Bearings for Radial and Thrust Loads.

Fig. 17 shows a radially loaded shaft held against endwise motion in either direction. This bearing is capable of carrying thrust load in either direction, but never more than one bearing on the same shaft should be held in this manner.



Figs. 19 and 20. Thrust Load in One Direction on Collar Bearings.

This bearing differs from the preceding mounting only therein that it has the outer race secured between shoulders in the housing. This arrangement and the preceding one are usually found combined on the same shaft, which is then held endwise at one point only, so that temperature changes, or deflections of the shaft can cause no cramping.

Fig. 18 shows separate radial bearings for radial and thrust loads. This type of bearing is used when it is desirable to take thrust load on bearings of the radial type, although the space available does not permit of a single radial bearing of sufficient diameter to take both loads. One bearing is then mounted entirely free circumferentially so as to take the radial load, while the other bearing is mounted between the shoulders, and takes the thrust load.

Figs. 19 and 20 show thrust loads on a collar bearing in one direction only. Here the stationary race is provided with a spherical seat so that it will distribute the load over the complete circle of balls. In order to permit compensating shifting, the fixed seat must be radially free of the shaft or of the housing. The shoulder on the shaft should be large enough so as not to permit any bending strains on the rotat-

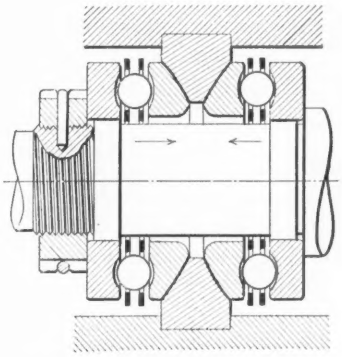


Fig. 21. Thrust Load in Two Directions on Two Collar Bearings.

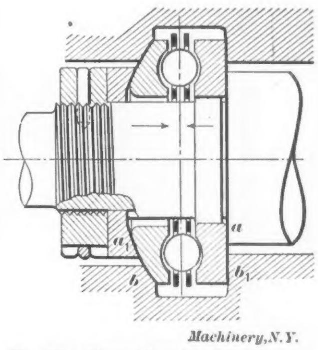


Fig. 22. Thrust Load in Two Directions on One Collar Bearing.

ing race. When it is inconvenient to provide a large enough shoulder on the shaft, a washer can be inserted between the shoulder and the race, as shown in Fig. 20. In Fig. 21 is simply shown a modification of the bearings in Figs. 19 and 20. This bearing takes thrust loads in two directions on two collar bearings. Fig. 22 shows an arrangement by which thrust load in two directions may be taken up by a single collar bearing. This arrangement is one which economizes space, cost of bearings, and number of parts. Fig. 23 shows an arrangement where a radial bearing is used for taking the radial thrust, and a collar bearing is used for taking the end thrust. Fig. 24 shows an arrangement for taking up radial load as well as thrust load in two directions, these loads being carried on one radial bearing and two collar bearings. In this design attention may be called to the distance piece inserted for binding the inner race of the radial bearing against the shoulder of the shaft.

It is occasionally inconvenient to arrange the bearing so that the parts of the inner races can be clamped to the shaft,

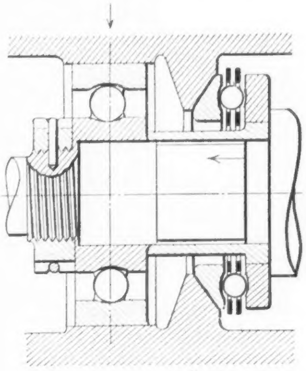


Fig. 23. Radial Load and Thrust Load in One Direction.

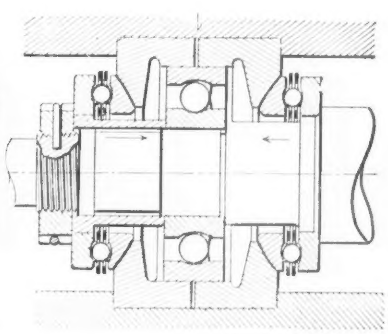


Fig. 24. Radial Load and Thrust Load in Two Directions.

or it may be desirable to have a shaft sliding through the bearing. In such cases a sleeve may be introduced on which the inner race of the bearing is firmly clamped endwise, the shaft simply resting in this sleeve. This gives a long bearing to the shaft, which would not be possible if the shaft was directly mounted in the ball race, because the peening effect of the vibrating loads would, even if the race itself, was prolonged, be concentrated on a narrow zone of the shaft. A bearing of this kind is shown in Fig. 25.

In Fig. 26 is shown a bearing which is intended for shafting which may not be fully to standard size. The inside of

the ball race is tapered, and a split bushing, tapered on the outside as shown, will, when tightened, bind the shaft as well as the race to it, and will compensate for all variations in size. The nut should not be used to draw the bushing in, but should merely act as a lock to hold it in place after it has been driven home with a soft hammer.

Ball bearings should always be enclosed so that lubricant will not be lost by leakage, and so that foreign matter will be excluded. Fig. 27 shows an efficient way of enclosing a ball bearing without using any packing. At the end where a shaft passes out of the enclosure, a flange should be bored out about 0.020 inch larger in diameter than the shaft. This flange should be separated into two lips by an angular groove, either cored or bored, as shown at *A*. These lips should not be less than 1/8 inch wide and should have sharp edges. The groove should be provided with a hole or narrow slot *B* at its lowest point to communicate with the bearing oil space. The groove itself should have a width of not less than 3/16 inch,

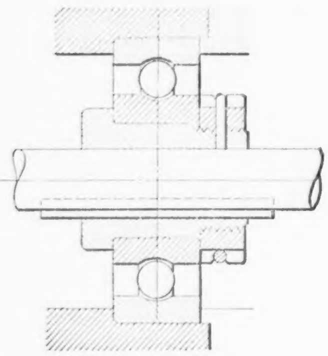


Fig. 25. Shaft Free in Longitudinal Direction in Inner Race.

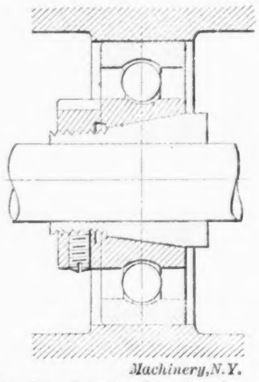


Fig. 26. Adapter Bearing for Shafting varying from Standard Size.

and a depth of about 5/16 to 3/8 inch, and should not be filled with packing material. Fig. 28 shows an arrangement of a similar kind, excepting that here is introduced a second groove and a third lip. This arrangement is employed where water may be occasionally encountered, and will prevent its entrance. What little may find its way past the outer lip into the outer groove is soon drained out again through the holes provided.

Where much grit is encountered, as in grinding machinery, a packing may be necessary, and filling the outer groove with a fairly consistent grease will provide such a packing without introducing friction. A bearing of this kind is shown in Fig. 29. A grease cup of the spring loaded piston type will automatically maintain the integrity of this packing. In some cases felt ring packings may be used, but these ought to be soaked in good soft paraffine, and a spring wire ring should be placed around the felt washer so as to force the

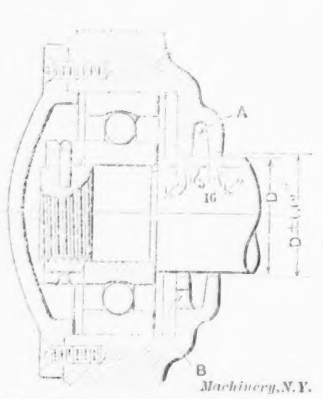


Fig. 27. Example of Enclosed Bearing.

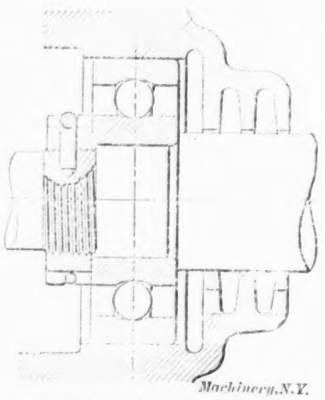


Fig. 28. Bearing Enclosure with Double Groove.

outer edges of the washer outward, which will cause the felt to come into more intimate contact on the sides. Felt washers may also be applied as shown in Fig. 31. Here the washer is tapered on one or on both sides, and the sealing of the enclosure is made entirely against the sides of the surfaces against which the felt washer bears. The felt washer is pressed inwards by means of springs on the outside. The two modifications shown in Figs. 30 and 31 are intended to be inserted between the faces of the stationary boss and the rotating hub. The modification in Fig. 32, however, is enclosed entirely within one or the other. A felt ring is set into a counter bore and held in place by a light metal cap sprung into position.

The author concluded his paper by saying that, on account of the time and space available at the society's meetings and in the society's transactions, much has been omitted in regard to ball bearings that might have been well considered, and

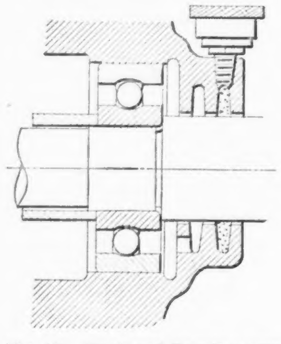


Fig. 29. Enclosed Bearing with Grease Packing.

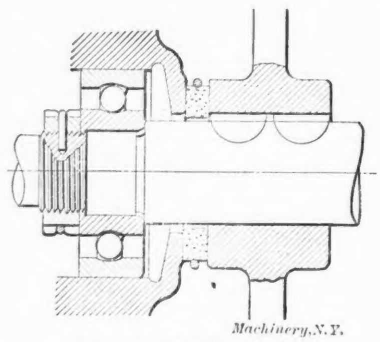


Fig. 30. Felt Ring Packing.

the author hoped in the future to make such contributions regarding the development of ball bearings as would appear to be of general interest.

* * *

In a paper regarding the economy attainable by the use of pneumatic tools, in preference to hand labor, for such operations as chipping, calking, drilling, riveting, etc., in British practice, Mr. C. P. Whitcombe states that the following comparison of the relative speed of turning out work by pneumatic tools and by hand labor has been compiled from actual results. It is not claimed that these speeds are applicable to all classes of work, but that they can be generally attained and sometimes even exceeded:

	Speeds.	
Description of Work.	By Hand.	By Pneumatic Tool.
Heavy chipping	1	2 to 4
Calking	1	3 to 4
Drilling	1	3 to 4
Rimering	1	4 to 6
Riveting	1 (3 men)	1½ to 2½ (2 men)

Figures recently prepared by a leading English firm for their own information showed that a weekly saving of more than $2 was being effected in wages for every $1 of running

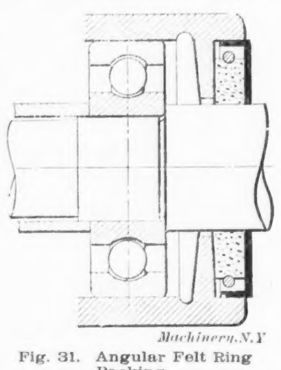


Fig. 31. Angular Felt Ring Packing.

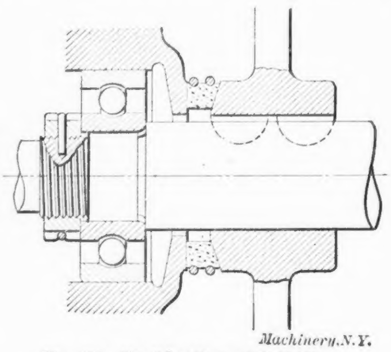


Fig. 32. Modification of Felt Ring Packing in Fig. 30.

cost of the pneumatic plant, the latter including interest on capital outlay, depreciation, power charges, maintenance and repairs.

* * *

In the article "A Draftsman's Tool Chest," by I. G. Bayley, published in the October issue, the fourth line of the second paragraph from the bottom of the second column should be changed to read: "An elastic band holds the door shut," etc. Also change "Instep's" tables on the following page of same article to "Inskip's" tables, etc.

* * *

A new example of the reliability feature of motor cars is given by the recent trials of a six-cylinder Hotchkiss car, which, according to the *Practical Engineer*, during its tests accomplished 21,250 miles, the longest trial on record, being 6,250 miles more than has been attempted by any other car. Of this distance, 10,474 miles was covered without a single involuntary stop.

TESTING THE DIVIDING HEAD OF THE CINCINNATI MILLING MACHINE.

The matter of inspecting work properly is a difficult problem to deal with, whether that work consist of parts of machinery, finished machines, drawings, calculations, or what not. There are so many ways in which a thing may be wrong that it is easily possible to neglect looking for one of the many in the course of a long day's work. Again, it is remarkably easy to follow another man's mistakes. Besides this, it takes remarkable strength of will to inspect as carefully and honestly at the end of nine or ten hours of work as at the beginning, or at the end of a year's work as at the commencement of it. The greatest aid an inspector can have in passing on the correctness of machine parts, machines or drawings is a regular laid out schedule telling him what to look out for, and establishing, in the case of machinery, the allowable limits of error. With a carefully planned schedule of this kind, the inspector can examine and check off each point item by item, and thus be able to do this work acceptably with no greater strain than the expenditure of that degree of care and intelligence which must be part of his equipment, if he is at all fit for his position.

The Cincinnati Milling Machine Co.

TEST SHEET FOR DIV. HEADS.

	Assembler's Check	Inspector's Check
Eccentric works freely	ok	ok
Spindle runs freely without worm	✓	✓
Split ring clamps spindle easily	✓	✓
Crank works freely with worm engaged	✓	✓
No shake in spindle	✓	✓
Index pins fit holes in index plates	✓	✓
No noise in gears, mitre gear bearing	✓	✓
Stop for side plate locks plate firmly	✓	✓
No back lash in gears, Limit ½ hole in side plate	✓	✓
Vernier flush with swivel block	✓	✓
No marred screws or oil cups	✓	✓
Block swivels 5 degrees below horizontal and 50 degrees beyond vertical	✓	✓
Smooth hole in spindle	✓	✓
Tail stock has zero line on swivel block	✓	✓
Elevating center has zero marks	✓	✓
Center in head has good bearing	✓	✓

	Maximum Error Allowed	Test in Thousandths	
Front taper hole true in spindle at outer end of 18″ test bar	.0015″	1	
Front taper hole in spindle true at nose	.00025″	1/4	
Spindle in line with tee slot in table at outer end of 18″ test bar	.001″	1	
		F	B
Spindle central with tee slot in table front or back	.002″	1	
Tail stock centers in line with Head center using zero marks — Large end	.001″		1/2
Tail stock centers in line with Head center using zero marks — Small end	.001″		1/2
Back taper hole in spindle true 4″ test bar	.005″	4	
Spindle perpendicular to table at outer end of 18″ test bar	.0015″	1	
Error in worm wheel. 8″ dia. test plate	.0015″	1/2	
Center runs true on point	.00025″	1/4	

Shop Order 3547 *Size* 10″ *Div. Head Number* 69-2887

Date Assembled Sept 5/07

Assembler's Number 311 *Name* R. Engel

Date Inspected Sept 18/07 *Inspector* H. Holz

Fig 1. A Sample Inspection Sheet for Cincinnati Dividing Head.

We illustrate and describe herewith the method pursued by the Cincinnati Milling Machine Co. in inspecting its dividing heads for workmanship and accuracy. The method of inspection pursued involves not only the regular schedule of investigation of which we have just spoken, but a carefully laid out series of inspection operations as well, with appropriate tools provided for the purpose.

The printed schedule is shown in Fig. 1. As may be seen, this is divided into two sections, the first of them dealing with items of fitting and workmanship which have to be checked by both the assembler and inspector; the second part gives a list of the maximum errors allowed in certain measurements and alignments, with a blank column in which the inspector records the results of his tests of these various dimensions. We are told by the manufacturers that this inspector's test sheet was drawn at random from a lot of fifty, it being the result of the inspection of a 10-inch dividing head examined on September 18, 1907. The inspection is made as nearly as practicable in the order listed on this sheet.

As previously stated, attention is first of all given to the workmanship on the different parts of the mechanism, each of which must come up to the required standard before the workman passes the head on to the inspector. At first glance, it might appear that these items need not be on the test sheets, since any good workman would naturally give these details his attention when assembling the head. However, experience has proven that it is best not to leave this to the workman's memory, it being better to remind him of the standard of excellence required and the test to be made on each of these parts by detailing each item on the test sheet. It is evident that the accuracy tests cannot be made until all of the mechanism has passed inspection.

For the tests relating to accuracy, various special tools are provided, with carefully worked out methods of procedure. The first test relates to the accuracy of the front taper hole in the spindle. Fig. 2 shows the operation with the special devices used. A test plate or table is provided, having a T-slot accurately made to standard size, into which the tongue of the dividing head fits. This table has a rearward extension provided with a knee the use of which will be explained later. In the front taper hole of the spindle is inserted a hollow test bar, with a taper shank carefully fitting the standard taper to which the hole is finished, and carefully made so that it is known to be straight, true to diameter and concentric, far within the usual limits of error in such work. The gage used is one of the American Watch Tool Co.'s test indicators—used on all their other accuracy tests as well. It is graduated to read to thousandths of an inch, and the graduations are so far apart that it is easy for an experienced workman to estimate halves and quarters of a thousandth. The shank of the hollow test bar having been carefully cleaned and lightly driven into the carefully cleaned taper hole of the diving head spindle, the indicator point is brought to bear on its outer end at a distance of 18 inches from the nose of the spindle. When the bar is revolved through the indexing mechanism, the error may be read direct from the indicator dial. The maximum error allowed is 0.0015 inch. In the present instance, the head showed an error of 0.001 inch, that is to say, 0.0005 on each side of the true position.

The second test determines the truth of the front taper hole by resting the indicator point against the side of the hole, the reading being taken in the same manner as before. The maximum error allowed on this test is 0.00025 inch, and in the present case the head tested within this limit.

The third test relates to the alignment of the spindle with the T-slot on the table. A little thought will show that in measuring this from the test arbor there is danger of introducing into the measurements the error of the untruth of the taper hole, previously determined, unless some provision is taken to eliminate that error. This is done very simply. First the test arbor is rotated with the indicator bearing on its side, until the "high point" is found. This position having been located, the spindle of the head is rotated 90 degrees, so that in Fig. 3 the "high point" would be at either the top or the under side of the bar. Under these conditions it is obvious that the two sides of the bar are for all practical purposes parallel with the axis of the spindle. The special standard on which the test indicator is mounted in Fig. 3 has a tongue which engages the T-slot of the fixture table. Readings with the indicator are taken at different points on one side of the bar. The difference between the largest and smallest of these readings gives the amount by which the arbor (and therefore the taper hole in the spindle) is out of alignment. In the present instance this error was found to be 0.001 inch in 18 inches. The maximum error allowed is 0.001 inch.

This is followed by a test to determine whether the center of the spindle is in the same vertical plane with the center of the T-slot. It is a repetition of the preceding test, except

that in this case the readings are taken at several points along both sides of the bar, the base in which the indicator is mounted being turned around in the slot to bring the measuring point on the other side of the spindle for this purpose. The readings on both sides for each position are noted and their difference taken; the difference determines the error. In the present instance there was an error of 0.001 inch in 18 inches, while a maximum of 0.002 inch is allowed.

Following this, comes a test to determine the alignment of the head- and tail-stock centers. This is shown in Fig. 4, and is a repetition in most respects of the test shown in Fig. 3, except that in this case an accurate test bar 12 inches long is held between centers. The tail-stock has a reversible center bar with a large center on one end and a smaller center adapted for the lighter classes of work of small diameter on the other end. Both of these are tested for alignment, and in the present case showed an error of only 0.0005 inch.

Since it is expected that the dividing head will sometimes be used with the work spindle clamped in any position between the limits through which it may be swiveled, a test is also made to determine whether or not the spindle swivels in a plane perpendicular to the top of the table. In this test the head is set so as to bring the spindle in a vertical position. The test bar is inserted in the spindle as shown in Fig. 2. The indicator is then mounted on a stand similar to that shown in Fig. 3. This is rested against the face of the vertical surface of the test block, and by taking several readings along the side of the test bar, the parallel relation of the bar with the face of the vertical surface of the test block is at once determined. In the present instance the error found was 0.001 inch, while the maximum allowed error is 0.0015.

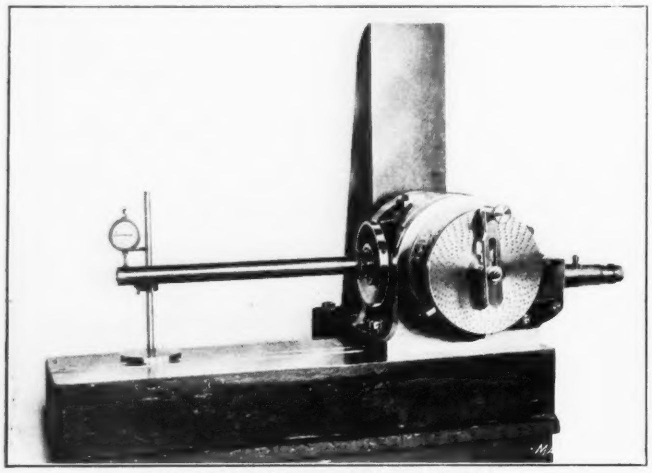


Fig. 2. Does the Taper Hole run True?

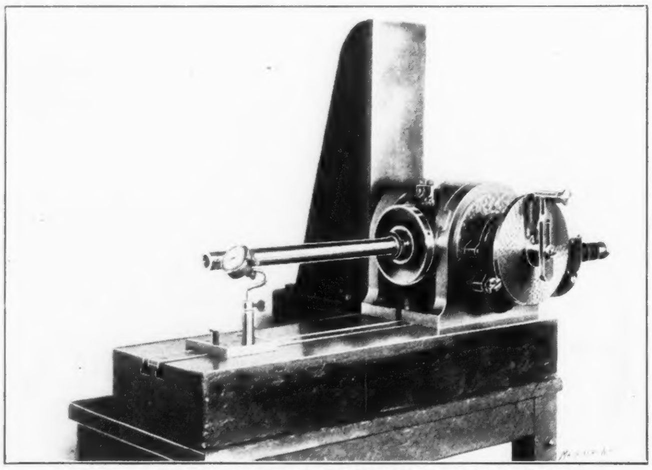


Fig. 3. Is the Axis of the Taper Hole True with the T-slot?

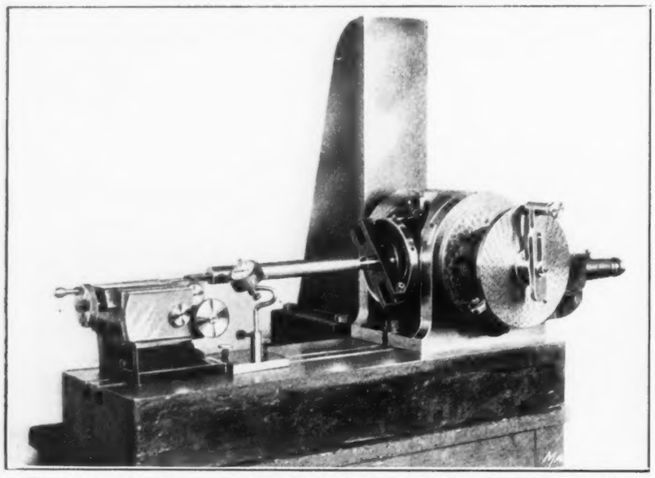


Fig. 4. Do the Head- and Tail-stock Centers line up with the T-slot?

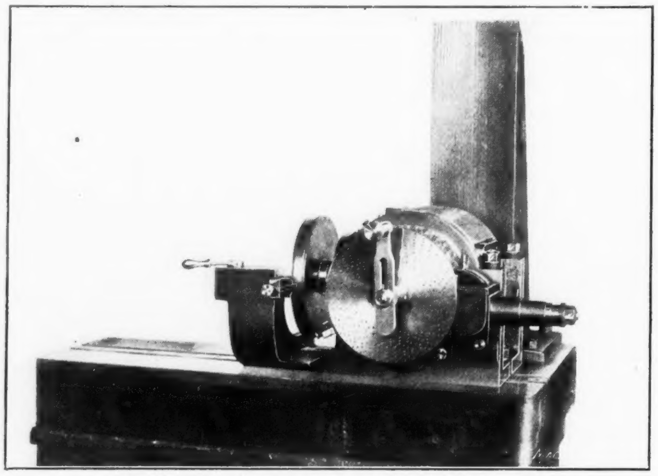


Fig. 5. How Accurate is the Index Worm-wheel?

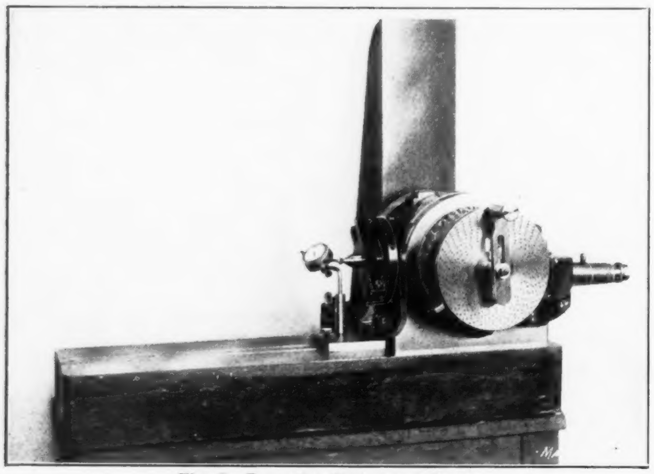


Fig. 6. Does the Center run True?

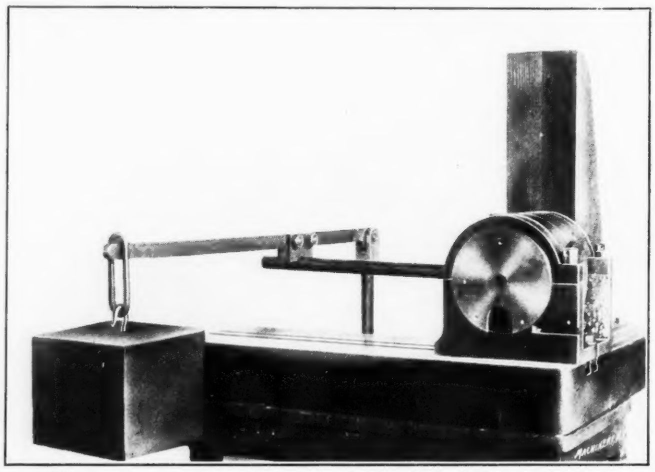


Fig. 7. Do the Clamp Bolts hold the Head firmly from Turning?

Perhaps the most ingenious and interesting of the tests is that used to determine the accuracy of the indexing. To test this, the apparatus shown in Fig. 5 is used. An arbor is placed in the taper hole of the spindle and on this arbor are clamped two disks, having nicely finished ground edges of the same diameter, matching each other so as to form an unbroken surface. To the testing plate is fastened a bracket carrying a marking device which, by the operation of the

small lever shown, scribes a fine line of uniform thickness across the edge of the two disks. The spindle is indexed for forty divisions and a line scribed after each indexing. Then the arbor holding the disks is removed from the dividing head, grasped in a vise, the clamp bolt is loosened, and one of the disks is rotated successively to seven different positions, each one-eighth of the circumference ahead of the preceding one so that in each case the first line scribed on it exactly matches the line on the other disk, corresponding with the selected position, and for each position the lines in the one-fourth of the circumference immediately ahead are examined for alignment under a strong magnifying glass. Their non-alignment indicates the error in the index mechanism, and may be estimated by an experienced man to very close limits. In the present instance the maximum accumulative error found in any one-fourth of the circumference of the disks was 0.0005 inch.

The above method of testing is frequently checked by the following plan: Instead of scribing hair lines across disks in Fig. 5, slots 1/4 inch wide and 1/4 inch deep are milled into them. Pains are taken to make these slots exactly to gage. Then the work is removed from the dividing head and the slots are compared by lining up any two of them the same way as in the previous test, except that in this case it is done by inserting a standard gage of sufficient length to fill both slots. The alignment of the other slots is then compared by testing with special gages. These gages are made 0.00025, 0.0005, 0.00075, 0.001 inch, etc., below standard, and obviously the amount that any given gage is below standard indicates the error or amount of non-alignment of those two slots into which it fits snugly.

Since accuracy in the other parts of a dividing head is of no great value when doing work between centers unless the head center itself runs true, this part is also tested, as shown in Fig. 6. This is a repetition of former tests, except that here the indicator point rests against the point of the head center, and when this is revolved through the index mechanism, the error may be read from the indicator. The maximum error allowed on this test is 0.00025 inch.

Besides these thorough inspection tests, others are made of the separate parts in the course of manufacture, and in the preliminary stages of the assembling. One of the most interesting of these, shown in Fig. 7, is employed for testing the rigidity of the clamping device by which the head is held at the proper angle. As shown, a bar 24 inches long is placed in the hole of the work spindle. A lever carrying a 200-pound weight on one end is then brought to bear on this bar at a point 22 inches from the swiveling center. The arms of this lever are in the ratio of 3 to 1, so that the total pressure brought to bear on the bar is 600 pounds, at a point 22 inches from the center of swivel. Every head must undergo this test without any evidence whatever of failure on the part of the clamps to rigidly hold the swivel block in position.

The dividing head is a most important part of the equipment of the milling machine, particularly in the case of the universal type. Its accuracy is trusted implicitly in many operations, and on its accuracy depends the accuracy of the work. We think it will be agreed that dividing heads that have successfully passed the tests outlined above are worthy of reliance in all ordinary operations.

* * *

There is very little literature on drop forging or "machine blacksmithing," which seems strange considering the importance of the art. The editor would gladly receive a good contribution on the subject. Any reader of MACHINERY who feels that he could write such an article will please communicate.

* * *

There are some curious misnomers in common use as, for example, "cork" legs, the name commonly applied to artificial limbs, which are not made of cork at all, but wood, the qualifying adjective being the name of the inventor. John Cork made the first wooden legs in the early part of the 19th century, his product being the first successful substitute for the "peg-leg." No artificial legs are made of cork, elm or willow being almost always used in preference to any other material.

TESTING THE LEAD OF TAPS AND SCREWS.

ERIK OBERG.*

In cases where there is no necessity of ascertaining the exact error in the lead of a screw or tap, and when only a limited number are to be tested, a fairly good test is afforded by simply screwing the thread into a female gage. The threaded portion of this latter should then, however, be fairly long, so that errors in lead, which are liable to be very small in a short distance, may be detected by taking account of the error in the comparatively long length. Ordinarily, however, when quantities of taps are to be tested, the errors in lead are most easily ascertained by some device particularly intended for the testing of the lead of a screw thread alone. Some devices which test both the lead and the diameter within certain limits are also in use. Of these latter, two examples are shown in a report on British standard systems for Limit Gages for Screw Threads, presented to the Engineering Standards Committee of Great Britain.

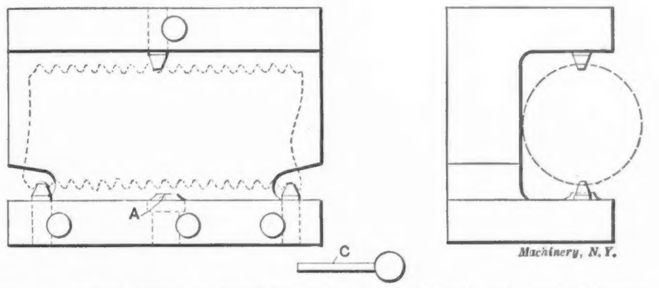


Fig. 1. British Gage for Simultaneous Testing of Lead and Angle Diameter.

Testing the Lead by Gages.

The first of these gages is shown in Fig. 1. In this gage, allowance is made for a permissible error in angle diameter and lead. As is plainly shown in the cut, the screw thread enters between three fixed points, shaped like the thread, two of which are located in the lower jaw of the gage, and one in the upper. The distance between the two points on the lower part of the gage should be equal to about twice the diameter of the screw. The fixed point in the upper jaw should, of course, be placed midway between the points in the lower jaw. At A is shown a ground flat face which is so adjusted that the small cylinder C, of such diameter that it will touch the thread about half way down its depth, will barely enter between the flat face and the thread of the bolt for the *minimum permissible* diameter, but will "not go" as a general rule. This device then gives a practical test for both diameter and lead. If the lead were out too much, the screw would not enter the gage, because the two points in the lower jaw would not fit the pitch of the thread, these points being, of course, set to a standard gage. If, again, it could be conceived that the diameter was so much smaller than the standard that the screw or tap could be placed in the gage in spite of the lead being an appreciable amount long or short, then the feeler C would enter so freely between the face A and the screw as to indicate that the screw was not within permissible limits. It will be noticed that provision is made for getting the points entering the threads placed exactly in the center of the screw. In the end view, the screw is indicated resting up against the back of the gage with one side, the distance from the back of the gage to the center of the points being equal to half the diameter of the screw. It is

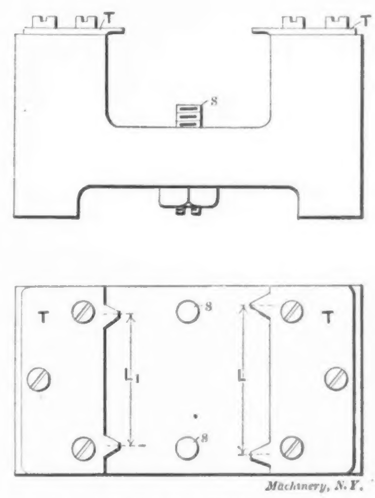


Fig. 2. Maximum and Minimum Gage for Lead and Angle Diameter.

* Associate Editor of MACHINERY.

evident that gages of this kind will have to be made for each separate diameter and pitch.

Another form of gage intended to deal with shorter lengths of thread than the one just described is shown in Fig. 2. In this case, two separate gages are applied, one minimum and one maximum. The screw is supposed to enter into the one and refuse to enter into the other. In this gage the top plates *T* are made of hardened steel and contain V-teeth set as shown, the distance *L* representing the next even number of threads immediately above the number contained in a length of screw equal to the diameter of the thread, while the distance L_1 is one thread shorter. The plates are screwed, and preferably doweled, to a base plate, and are, of course, made and adjusted to a standard plug. At *s* are shown screws which can be so adjusted that the measurement can be made exactly at the center of the screw, the distance from the faces of screws *s* to the center of the gage plates being equal to one-half the diameter of the screw.

Comparators for the Lead of Taps and Screws.

When it is wanted, however, to determine the errors in pitch with some exactitude, and not only to find out whether the error is between certain limits, then the instrument termed thread comparator is used. This consists, in its simplest form (see Fig. 3), of a fixed block *A* and a sliding block *B* provided with ball points. The sliding block operates a pointer *C*, which on a large scale indexes the errors of lead. The manner of using this instrument is as follows. A standard plug is first placed against the device so that the ball points enter in threads, say one inch apart. The position of the pointer on the scale is noted when the standard plug engages the ball points, the free block *B* adjusting itself to the thread into which its ball point enters, and carrying with it the pointer *C*. Next, the tap or screw to be tested is placed in position against the device. If the lead of this screw or tap is correct, and is the same as that of the plug, the pointer will evidently occupy the same position in relation to the scale as in the case of the plug. If the tap or screw is long or short in the lead, the pointer will show the amount on the scale by swinging either to the left or to the right. The scale should, of course, preferably be graduated so as to show thousandths of an inch.

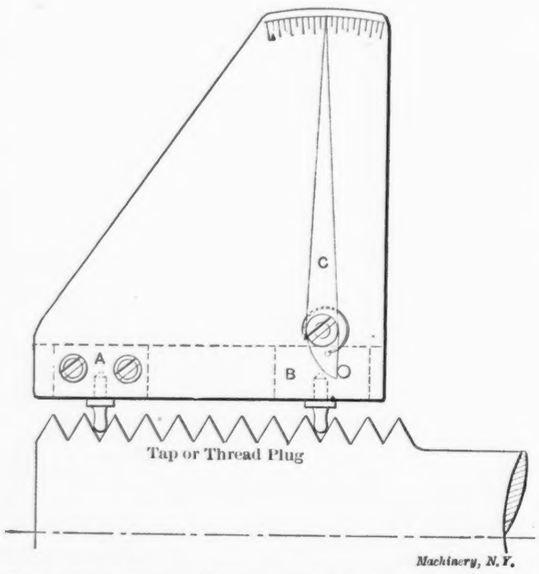


Fig. 3. Simple Form of Comparator for Lead of Screw Thread.

A more elaborate device for measuring the errors in lead of taps is shown in Fig. 4. Here one ball point *A*, which we may call the fixed, is mounted in a slide *D*, which latter is operated by a knurled head screw *B*. Ball point *A* may be screwed into any of the holes *C*, which may be ½ inch apart; thus one may, with this device, measure the lead in one inch, or in any length up to six inches, as may be desired, by moving the ball point *A* to different positions in the slide *D*. The ball point *E* is inserted in a movable block resting on a ball bearing. This block, in turn, is connected through the lever *F* with the indicator or sensitive gage *G*, which should be so arranged and graduated that each thousandth inch can be easily read. When the standard plug is placed against this device, the ball points entering between threads in the same way as in the former device described, the slide *D* can be so adjusted by the knurled head screw *B* that the indicator points to zero. When the screw or tap to be tested is placed against the ball points, any error will then be apparent by the motion imparted by too long or too short lead to the movable ball point *E*. This motion, of course, is, through the lever arm *F*, carried to the indicator. If the latter is graduated in thousandths of an inch, the graduations below or above zero will indicate the amount in thousandths of an inch that a tap or screw is short or long in the lead in the distance originally measured on the plug, *i. e.*, the distance between the ball points when the plug was placed in position against the device. In the device shown, the length of the lever *F*, between its pivot and that end which is operated by the movable block, is half of the length between the pivot and the end operating the gage. Consequently, if the gage be graduated to show movements of 0.001 inch on its own plunger, it will indicate a motion of 0.001 inch on the movable ball point by moving two graduations on its own scale. Very close measurements are consequently possible.

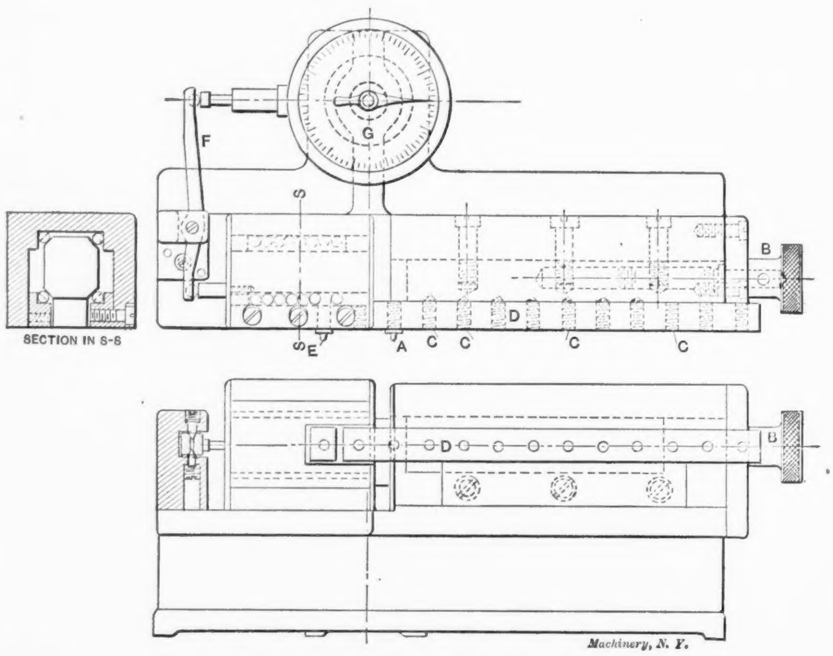


Fig. 4. Comparator for Testing Lead of Taps and Screws.

Of course, this device is only one modification of the many possible for obtaining the same results. Very likely there are others equally good, but this one is shown as an example of a satisfactory design, and at the same time as an indication of the principles involved in the design of comparators for the lead of screw and tap threads.

* * *

MACHINERY'S REFERENCE SERIES.

In the advertising section of this issue, announcement is made of a series of valuable reprints from MACHINERY. During the past fourteen years a large number of articles on machine design and shop practice has been published in its columns, all well worth preservation in permanent form. A number of well-known books first appeared by installments in MACHINERY, but these represent a comparatively small part of the valuable technical material that has appeared therein. The remainder represents the shorter contributions of many designers and others connected with the best constructive work. In order to make it generally available, we have selected the representative articles and divided them into groups, each group devoted to one subject, which will be published in 6 x 9-inch pamphlets, ranging from 32 to 48 pages. These pamphlets will contain only such articles as are considered to be of the most practical use, and they will be edited and condensed so as to be "all wheat."

LETTERS UPON PRACTICAL SUBJECTS.

REDUCING THE DIAMETER OF TOOL STEEL UNDER THE STEAM HAMMER.

It is often necessary to reduce the diameter of a piece of tool steel from its original size to perhaps one-half that diameter. This is very common in the making of taps and reamers of large diameters, where it is wanted to have the shank of considerable smaller diameter than the main part of the tool itself. It is evident that it is a great deal cheaper to forge down the diameter of the shank in such cases, than to use solid stock of the full diameter of the tool, and reduce it by turning, but it is necessary that the work of reducing the diameter under the steam hammer is done in the proper manner. Many blacksmiths do not seem to know how this work should be properly accomplished. I have seen many of them take a bar of steel, put it into the fire, leaving it there until the bottom side had arrived at a red heat, and then turn it and leave it in the fire until the other side got heated, paying no attention to the uniformity of the heat of the piece. The work is then taken to the steam hammer and reduced by continually rolling it around on the sides until it is reduced to the size wanted.

The result of this procedure is always a forging with a spongy or "piped" center. When this sponginess is finally

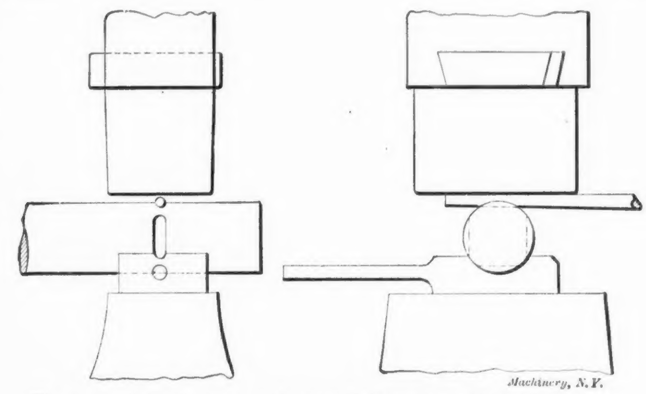


Fig. 1. First Operation in reducing the Diameter of a Tool Steel Bar under the Steam Hammer.

detected in the tool, the steel is blamed as being poor, but as a matter of fact, in most cases, the steel has been satisfactory to start with, and the fault is to be found in improper treatment in the blacksmith shop.

The proper way to reduce the diameter of a piece of tool steel is to first heat it uniformly, and then place it in the steam hammer as shown in Fig. 1. The blacksmith then proceeds to mark the bar on all four sides with a $\frac{3}{4}$-inch round machine steel bar, long enough to hold by the hand. This marking is intended to give a guidance as to the amount of reduction necessary. When the four sides have been marked

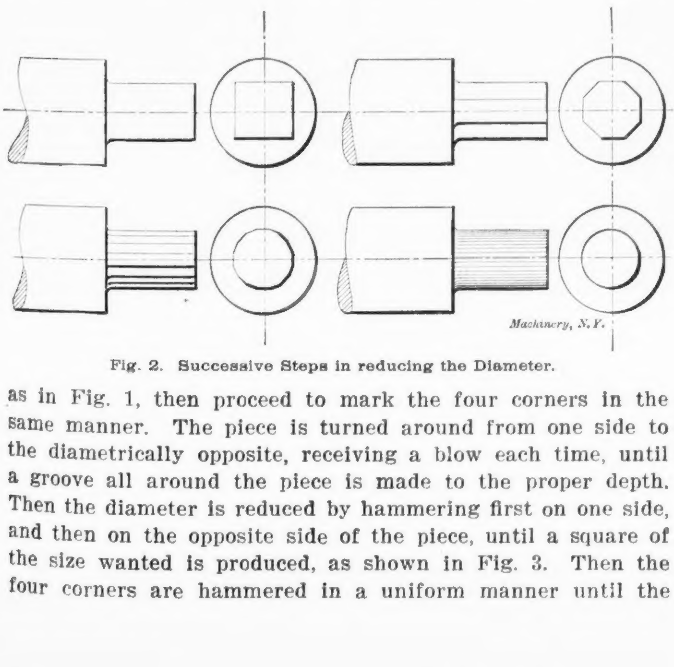
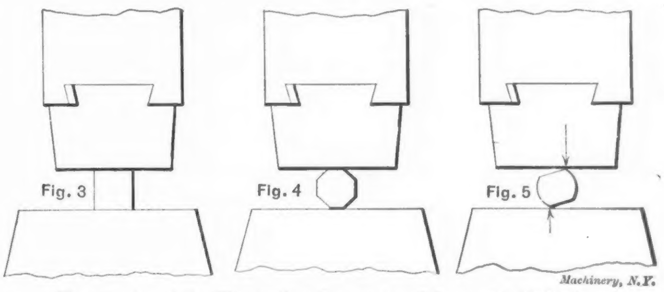


Fig. 2. Successive Steps in reducing the Diameter.

as in Fig. 1, then proceed to mark the four corners in the same manner. The piece is turned around from one side to the diametrically opposite, receiving a blow each time, until a groove all around the piece is made to the proper depth. Then the diameter is reduced by hammering first on one side, and then on the opposite side of the piece, until a square of the size wanted is produced, as shown in Fig. 3. Then the four corners are hammered in a uniform manner until the piece gets an octagon shape, as in Fig. 4. Next the eight corners of the octagon are hammered down, making sixteen sides, always making sure that the next corner hammered down is diametrically opposite the one just operated upon. Finally, if a swage of the proper size is on hand, the piece can be rounded with this; otherwise when all the corners have been reduced so that they are hardly visible, it is possible to round the piece nicely with even hammer blows until



Figs. 3, 4 and 5. Illustrating Corrrect and Incorrect Methods of reducing the Diameter.

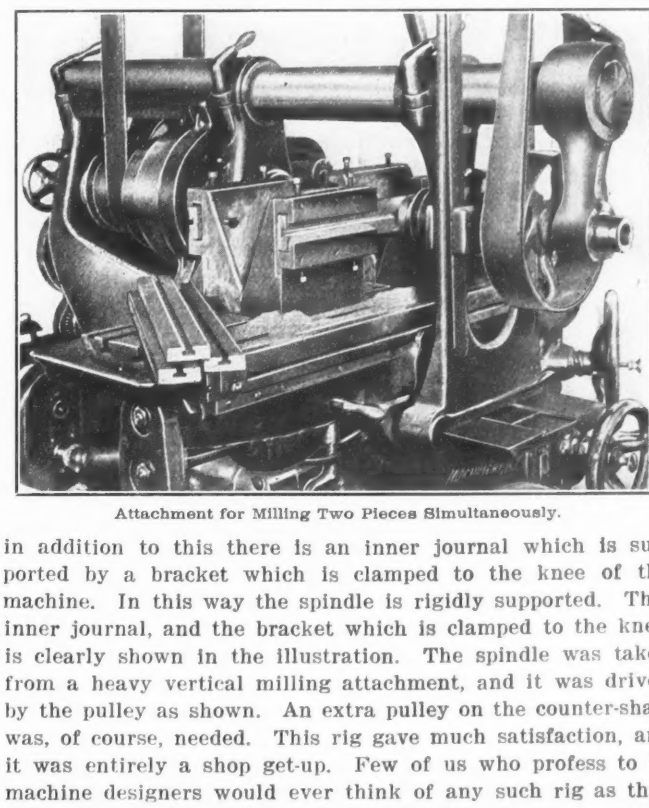
the correct size is arrived at. In Fig. 2 are shown the consecutive shapes assumed by a piece of steel worked down in the manner described.

It is evident that by rounding continually after the first blow is struck, the blow, as shown in Fig. 5, is not directed on a point that has a firm support directly under it, and a kind of twisting action takes place, causing one-half of the bar to have a tendency to slide in relation to the other half of the bar, the result being that the center of the bar is spoiled, and a spongy or, perhaps, a piped center results. In some cases this hollow center is of no consequence, as, for instance, when a hole is drilled through the piece, but it is evident that all efforts should be directed to avoiding results of this kind.

Decatur, Ill. GEO. T. COLES.

DOUBLING THE CAPACITY OF THE MILLING MACHINE.

The half-tone shows an improvised duplex milling machine. The machine, a Becker-Brainard No. 3, was provided with two arbor supports so designed that the arbor passed through them allowing the supports to be adjusted close to the cutter. The extra spindle is journaled in the outer arbor support, and



Attachment for Milling Two Pieces Simultaneously.

in addition to this there is an inner journal which is supported by a bracket which is clamped to the knee of the machine. In this way the spindle is rigidly supported. This inner journal, and the bracket which is clamped to the knee, is clearly shown in the illustration. The spindle was taken from a heavy vertical milling attachment, and it was driven by the pulley as shown. An extra pulley on the counter-shaft was, of course, needed. This rig gave much satisfaction, and it was entirely a shop get-up. Few of us who profess to be machine designers would ever think of any such rig as this. The man on the job simply had so much work of a certain

class to do, that he sought for some way of doing the work more rapidly, and this attachment was the result.

The work, as can be seen, was that of milling tee-slots in strips which were used for holding trip dogs. These strips were about 20 inches long. The fixtures for holding the work are also worthy of mention. They are simple, but effective, as they hold the work securely. Because of their simplicity, they are easily operated, and they are not likely to get out of order. The work which is not finished on the edges, is milled on the top and slotted. The fixture is so arranged that this milled surface is used to line the work up sideways. The screws, which are shown in the illustration, are used to adjust the work lengthwise until the slot is parallel with the table. This work might be done on a vertical machine, but there is an advantage in having it in the position shown, as the cutter can more readily free itself of chips. This advantage is more marked when the cutter is fed in such a direction that it cuts into the metal from underneath so that the teeth carry the chips up over it and drop them in its wake, instead of continually dragging them into the cut, as would be the case should the cutter run over and into the work.

Work, other than that shown, might be done to advantage in this way, as the fixture is quite inexpensive even when made specially for the work, the only parts necessary outside of the regular equipment, being the spindle and pulley. I trust that some of the readers of MACHINERY have been looking for something of this kind; if not, it may suggest something else which will prove to be a time saver, and give just as good satisfaction. JOHN EDGAR.

Hyde Park, Mass.

[A similar scheme for doubling the effective capacity of small milling machines was illustrated in the article "The Manufacture of Colt's Automatic Army Pistol," May, 1906, issue.—EDITOR.]

NOVEL METHOD OF PRODUCING THREADS.

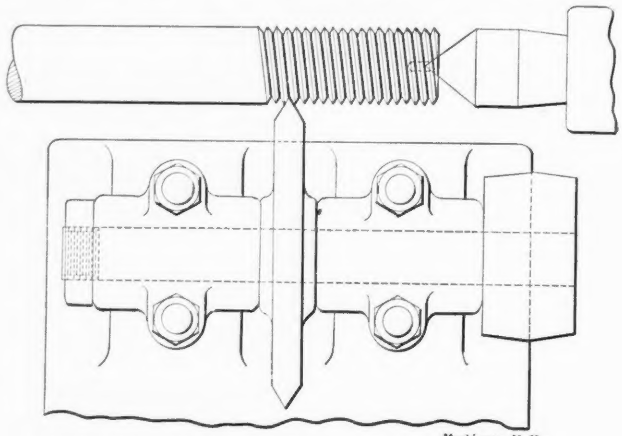
The accompanying cut illustrates a method used for threading studs, pins, etc., of manganese steel, this material being so hard that it cannot be cut by any kind of tool steel. A plain, hardened tool steel disk, having the edge made according to the angle of thread, is employed. This disk is revolved at a high speed, and at the same time forced into the work, which is revolved slowly. Due to the friction between the edge of the disk and the work, and the softening of the material, owing to the heat generated by the friction, the disk wears away the stock, and, by means of this, creates the thread. The stock is coming off in very small, thin scales like chips, which to some extent remind one of the scales of a fish. An ordinary lathe has been rigged up for the purpose of removing the tool-post and top-rest, and substituting for them the fixture shown in the cut. The disk must be driven independently by an overhead drum, or some similar arrangement.

The peripheral speed of the disk is usually between 3,000 and 4,000 feet per minute. The operation is unavoidably slow and expensive, and the method is used only when no other way is possible. The writer thinks, however, that the efficiency can be increased to some extent by increasing the peripheral speed of the disk, perhaps as high as 24,000 feet per minute, same as used on friction saws.

It is likely that high-speed steel would be preferable to ordinary tool steel as material for the disks, but, as the process described is necessarily slow, and is used only when no other way of threading is possible, it has not as yet been developed to the limit of its capacity. There is a certain point in the gradual development of the method above which it becomes economically preferable to employ high-speed steel for the disk, but below this point of development, although high-speed steel may be the best, the ordinary tool steel disk, owing to its smaller first cost, is economically the one to be preferred. A preference for the one or the other kind of steel is influenced by a number of factors, *viz.*, the number of pieces to be threaded per unit of time; the peripheral speed of the disk; the pressure between disk and work; and the efficiency of the system of cooling.

The question of cooling is in itself an interesting one. The reason why the heat does not draw the temper of the tool steel in the disk while the heat is so great that it softens the metal of the work, is that the disk is revolving at a high speed and the work only revolving very slowly, so that a unit of length of the periphery of the disk is in contact with the work but a very short time, while every point on the work, at the place where it is cut, is in contact with the disk a comparatively long time. Owing to this the disk has ample time to cool off, while the work accumulates the generated heat. The high speed of the disk also throws the film of air nearest to the disk outward, owing to the centrifugal force, and new cool air comes constantly in at the center, a current of air thus at all times tending to cool the disk.

The cooling thus obtained is found to be satisfactory at the present speed at which the disk is run, but at a higher speed



Cutting Thread by a rapidly revolving Hardened Steel Disk.

a system of cooling by an air jet, or still better, perhaps, by water, could be employed to advantage. This would also increase the limits within which an ordinary tool-steel disk could be used to advantage. For increasing the peripheral speed of the disk as previously mentioned, undoubtedly the best way would be to increase the diameter of the disk, permitting the number of revolutions to remain the same as before; but at the present stage of the development of this device there are some limitations to the size of the disk, inasmuch as it is used in an ordinary lathe, and the space possible to utilize for the disk is not very great. Another difficulty in increasing the diameter, rather than the number of revolutions, is that for a large diameter disk it is necessary to arrange the disk on an inclined angle in relation to the work in order to get a perfect thread, and this necessarily means a more expensive rigging.

The principle involved in this method of cutting threads is the same as that involved in the friction saw. But the principle of the latter machine cannot be carried out to its full extent in the present case, because the steel to be threaded must not be heated more than to a certain degree. Above this limit, increased heating would mean injury to the quality of the steel. The heat also must not be so high that it burns the thread.

If the call for threaded parts of this kind of steel would be great enough so that a special machine would be warranted to be built, then the efficiency of the method could be increased by a careful taking care of all the points previously referred to, but, at the present time, the demand is not large enough to warrant the expenditure of building such a machine.

High Bridge, N. J. OSKAR KYLIN.

TEMPERING DROP FORGING DIES.

It has always been a source of wonder to the writer, why so many drop forging dies are cracked in hardening, because it is not at all necessary that this should happen. Uneven heating, and uneven cooling with consequent uneven contraction is the cause of the trouble.

If drop forging dies are made from machine steel, they should be packed in No. 1 raw bone and fine wood charcoal, three parts charcoal being used for each two parts raw bone. They are then heated in an oven for eight hours, at a temperature of 1600 degrees F., and are then dipped the same as

described in the following for tool steel. When the dies are made of tool steel, let us first notice that the heating of dies or any kind of tools made from high carbon steel in an open furnace, even if covered with coke, is very injurious to the steel, especially so in the case of drop forging dies, as the carbon leaves the surface of the steel, and the dies will not harden on the outside, but will be harder further in. This

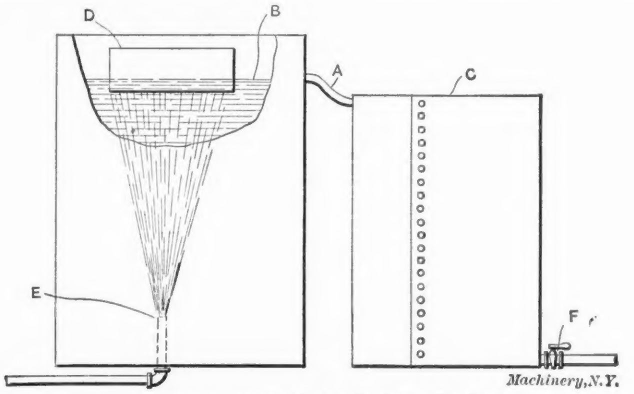


Fig. 1. Arrangement of Brine Tank for Hardening.

does not matter so much with tools that are to be ground to size after hardening, but it is a poor practice with any kind of tool-steel tools. Tool-steel dies should be packed in fine wood charcoal in a box large enough to allow plenty of charcoal between the die and the box walls, say about two inches or more. Seal the cover on tightly with asbestos cement, place the box containing the die in the furnace, and, if a pyrometer is attached to the furnace, hold the furnace at about 1,500 degrees F., leaving the die in for at least four hours. For a small die, shorter time will be sufficient, but a die weighing 50 pounds or more should be allowed four hours to heat slowly and uniformly. Then, instead of dumping the

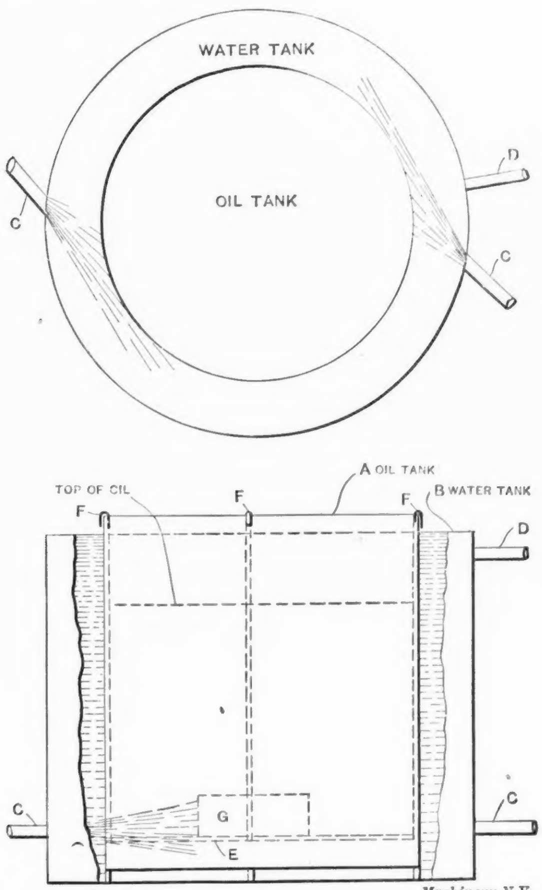


Fig. 2. Oil Tank for Hardening Room

whole die in a tank of cold, clear water, have two tanks, a large one and a smaller one, as shown in Fig. 1. An overflow pipe or hose *A* from the water line *B* in the large tank should connect it with the small tank *C*. When ready to dip the die *D*, place the face only in the water. Plenty of salt should be well dissolved in the water, about 4 pounds to the gallon; this extracts the heat from the die quicker than clear water, and prevents steam formation on the face of the die. A water pipe *E* should be carried in at the center of the large tank at the bottom, and should be supplied with water at fairly high pressure. When placing the die in the bath, open the valve of the pipe *E*, thus forcing the cold solution against the face of the die, while the warm water passes into the smaller tank. The solution collecting in the smaller tank, when cool enough, can be used for smaller tools, and, when so desired, can be run off by outlet *F*. Another bath, in an oil tank, inside of a water tank, as shown in Fig. 2 should be provided. The size of the tanks must be determined by the size of the dies to be hardened. Fish oil should be used in this latter tank, and the tank should have two water inlets *C*, at opposite sides of tank, and so arranged as to allow water to flow around all sides of the oil tank as indicated in the plan view. Pipe *D* is the overflow. A coarse mesh sieve *E* is suspended in the oil tank, and held by rods *F*. The oil tank should have four legs about 6 inches long, to allow water underneath the tank. When the die face has been cooled in the salt water solution, remove the die quickly to the oil tank, and lower it until it rests on the sieve (see *G*, Fig. 2). Let the die remain in this position until cold. It requires no further attention than removal from the oil. Dies hardened in this manner will not crack.

Lansing, Mich. J. F. SALLOWS.

BORING DRIVING BOX BRASSES.

The jigs illustrated in the accompanying cuts are used for holding driving boxes while their brasses are being bored. It is not claimed that these jigs, or the methods of boring boxes, which will be described, are the best in use, still they are preferable to the tools and methods employed in many shops for doing this work. Fig. 1 shows a jig which is used on a vertical boring mill. This jig is composed principally of two castings in the form of angle plates. The distance between these angle plates, or jaws, is made to suit the largest box in use, and the jaw to the left is made adjustable so that the jig can be used for the smaller boxes. When the jaw is to be adjusted, the four cap-screws which hold it are

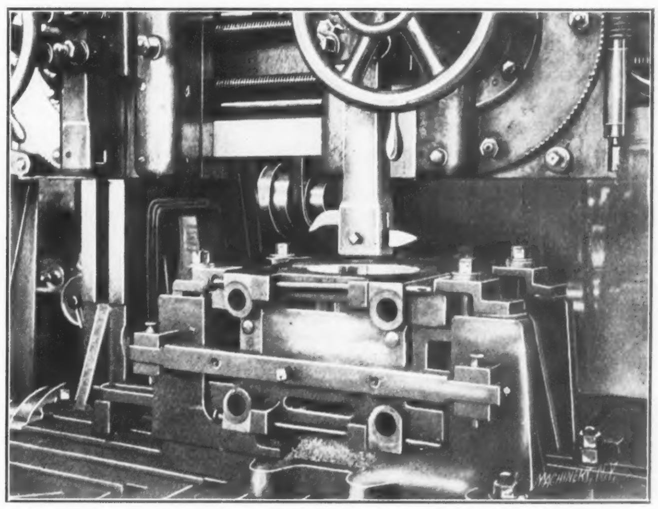


Fig. 1. Jig for Boring Driving Boxes on a Vertical Boring Mill.

removed, the angle piece moved in, and the cap-screws inserted into another set of holes which were previously drilled and tapped in the proper place. The jig is provided with two tie bars, one across each end. These tie bars, one of which is shown in Fig. 1, are for the purpose of taking the end thrust when heavy cuts are being taken, and they also facilitate the adjustment of the box. The forged lugs, which hold the tie bars, are held in place by nuts, which are on the inside of the jaws. A shoe, which can be seen in the illustration, is fastened to the right jaw. The shoe faces of all driving boxes are clamped against this shoe. When a set of boxes is to be bored, one is first laid out central with the shoe and wedge faces, then this box is clamped in the jig and the jig is set by it. All the boxes are then bored without shifting the jig. The roughing and finishing tools are, as shown in Fig. 1, both clamped in one tool-holder, and the roughing

and finishing cuts taken simultaneously, one cut usually being sufficient to complete the operation.

Fig. 2 shows a jig which is used when boring driving-box brasses on a horizontal boring mill. A detail of this jig is also shown in Fig. 3. The set-screws *A* provide means for adjustment, and also help to hold the box in place. The shoe face of the box is clamped firmly to the face *B* on the jig by bolts which fit into the tee slots *D*, and by a clamp which passes across the wedge face of the box. These bolts and the clamp are shown in Fig. 2. When a narrow box is to be bored, it is set central with the face *B* on the jig by

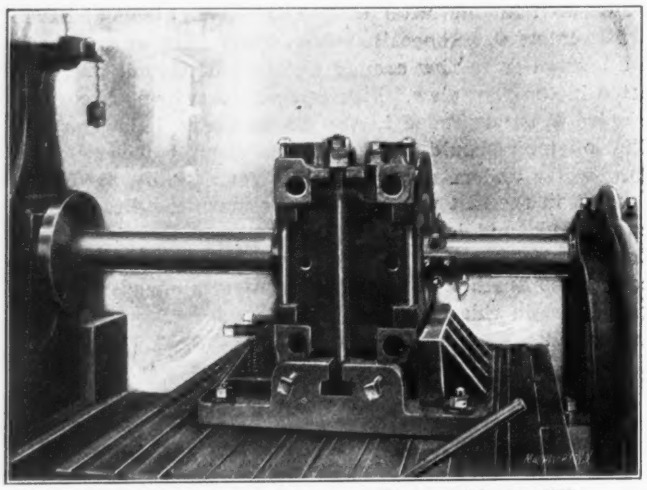


Fig. 2. Jig for Boring Boxes on a Horizontal Boring Mill.

bolting a parallel piece along the face *C*. This jig is set by practically the same method described for the vertical mill. One box is first laid out central and clamped in the jig, the jig is then set on the table, and the table set to the proper height and clamped. All the boxes are then bored without altering the position of the table. In this way, the distance from the bore to the shoe face of each box is exactly the same. Many prefer to set up each box separately, but I have found

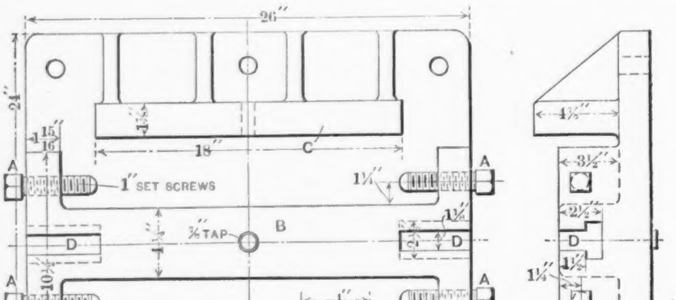


Fig. 3. Detail of Jig used on Horizontal Boring Mill.

the method described to give the best results. This boring bar also contains two tools, each tool having an adjustment which saves much time in setting the tools.

Port Huron, Mich. M. H. WESTBROOK.

THE MARVELOUS CO_2 ENGINE.

Inventive genius cannot be suppressed, not even, or particularly not, in Philadelphia. Time and again has that city seen the birth and death of the most stupendous labor-saving inventions. Though its inventors have not exactly hitched their car to a star, they have harnessed nature's forces and confined them to new and fearsome ways to do their bidding. It it not so long ago that we had Keeley with us, and not a few still lament his demise before he was able to impart the full secret of the connection between the interstellar vibrations, his fiddle, and an engine that would endlessly produce power after Keeley drew his bow.

But in the drawing of the big bow worthy successors have arisen. The confiding public is asked to subscribe to the stock of a development company, whose purpose is the creation of sub-companies, that are in turn to run engines with carbonic acid gas. The development company asks only the trifle of $10,000,000 in subscriptions. Just what is to be done with the $10,000,000 is not quite clear, unless it be presumed that the projectors consider the difficulty of inducing participation in the sub-companies such as to demand such payment. But hold! The prospectus does say: "The company has purchased from the inventor the sole ownership of the invention and all foreign and American patents pertaining to same." Possibly the company is signalizing its wide departure from nature's laws by a similar departure from the usual ones of greedy high finance, and donating a substantial portion of this $10,000,000 to the inventor as a small recognition of his genius.

One of the sub-companies to be formed is for automobile development in adapting the carbonic acid gas engine (CO_2 engine sounds more mysterious, and so spells greater possibilities) to the automobile. Who would not immediately mortgage his house, his salary, and his socks to buy a few shares at par, particularly when informed that "Last year, in the United States alone, over 160,000 automobiles were sold. If this company derived a royalty of but $100 per machine, we would earn $16,000,000." That the royalty of $100 per machine is very modest indeed must be clear, when it is realized that "an automobile will be reduced to about one-quarter of its present weight." In the automobile of to-day, the gasoline motor (so soon to be relegated into the limbo of forgotten toys), with the change gear box and radiator does take up somewhat less than three-fourths of the weight of the car. It follows that the CO_2 engine must weigh some more or less indefinite amount less than nothing. Surely a royalty of only $100 for a motor weighing less than nothing is a very modest one. Consider that a weight of less than nothing is a negative gravity, that one of our noted scientific novelists showed some years ago that a box of "negative gravity" worn on the back of its owner would set him afloat in the air, and the CO_2 company would seem to have solved the problem not only of the automobile but of aerial navigation as well, at one fell swoop. If there is any gratitude still extant in these degenerate days, the company cannot do less than immediately double its capital stock and issue half to me as a modest recognition for pointing out this unsuspected fact.

Does anybody still doubt? Hear the company: "A group of well-known, practical expert engineers in the city of Philadelphia has witnessed a demonstration of our generator and have stamped it with their approval." Are these expert engineers mere men of theory, whose investigations and opinions the world has so far been deluded into accepting as having weight? Not a bit of it! It is "the most scrutinizing inspection of practical men" that is wanted, not the preconceived notion of theoretical faddists wedded to foolish belief in accepted natural laws. "The layman can have little hesitancy" after hearing a report from one of these practical scrutinizers: "I have no doubt—not the slightest—the CO_2 generator does all that is claimed for it. I have seen the engine with the invention attached and studied it, and failed to find a single defect. Why it does it, of course, I don't know—its possibilities are limitless."

Does the company want to secure for itself all of the filthy lucre derivable from these "limitless possibilities?" No! In a noble manifestation of altruistic spirit it prefers to let the others who will form the sub-companies garner the gold. All it wants is $10,000,000 for granting this privilege to these fortunate sub-companies. Oh, proud Philadelphia, potent indeed is the spirit of brotherly love within thy borders when even the exploiter of such marvels (or shall we say marvelous exploiter) can so share with others!

A printed pamphlet with beautiful illustrations of a CO_2 automobile, of a CO_2 100 H. P. engine, of a CO_2 railway car, and

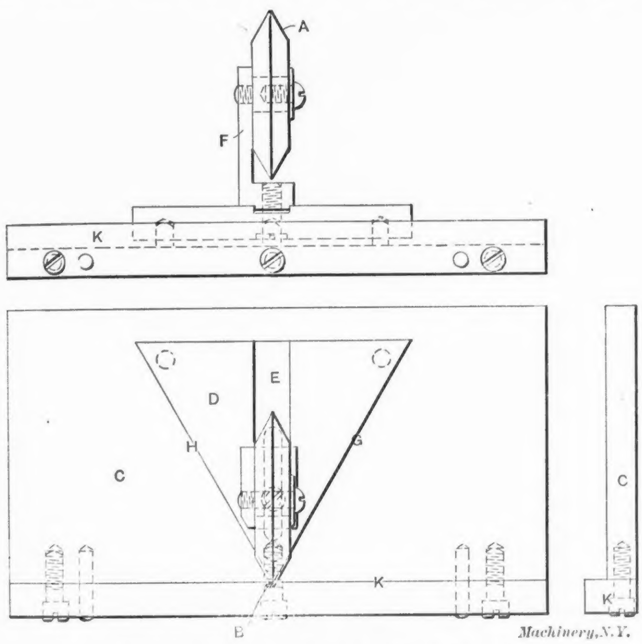
of a residence equipped with a CO_2 system can be had on demand—not a penny, not a postage stamp even, is asked in payment.

HENRY HESS.

Philadelphia, Pa.

FIXTURE FOR GRINDING ANGULAR MILLING CUTTERS.

The accompanying cut shows a little device which has proved itself very useful in grinding angular milling cutters when a perfect angle is required. The radius at the point of the angle can also be ground, radius and angle being ground at one setting. This fixture consists of a base-plate *C*, which is clamped to the grinder table so that it can be fed to and from the wheel by the feed arrangement on the grinder. On this base-plate rests a triangular plate *D*, carried on three feet. This latter plate is free to move in all directions, simply sliding on its feet on the plate *C*, and is guided only by the hands of the operator. In this triangular plate *D* there is a slot *E*, into which a tongue of the bracket *F* is fitted, this bracket then being movable back and forth on the plate *D*, and having arrangement for clamping in any position. The cutters *A* are clamped to this bracket *F* by a suitable screw and washer. For different widths of cutters, either different



Fixture for Grinding Angular Milling Cutters.

brackets must be employed or washers may be interposed between the bracket and the cutter, because it is evident that the center line of the cutter must always coincide with the center line of the triangular plate *D*. The cutter can be set to any given radius between the two angular faces by placing a gage block, having the same thickness as the radius wanted, against the point *B* of the triangular block, and placing a square against the gage, and adjusting the cutter so that the blade of the square just touches the point of the teeth of the cutter *A*. If I, for instance, have a cutter that I want to grind to a 60-degree angle, and want 1/16 inch radius at the point, I simply set my cutter central with the triangular block, and place a 1/16-inch gage block between the square and the point *B* of the block, and then adjust the cutter until it touches the blade of the square. The cutter is then clamped in place. The grinding itself is performed by sliding the plate *D* first to one side and then to the other, so that the sides *G* and *H* alternately rest up against the guide *K* on the bed-plate *C*, the side of the teeth of the cutter being meanwhile moved back and forth across the face of the grinding wheel. The turning around of the triangular block from one side to the other with the point *B* against the guide *K* evidently produces a radius at the point of the cutter between the two angular sides. The height of the cutter tooth in a horizontal direction, when setting, is determined by a gage block of such a height that the tooth face is in a horizontal plane with the center line of the cutter. The cutters are formed closely, before hardening and grinding, to the desired shape, so that there is but a few thousandths inch left to be removed when grinding.

P. YORGENSEN.

Hartford, Conn.

[The manner of determining the radius to be ground is not quite correct. When a gage block 1/16 inch thick is used, as described by our correspondent, the resulting radius of the cutter is not 1/16 inch, but, for a 60-degree cutter, having a 30-degree angularity on each side of the center line, only one-half of 1/16, or 1/32. If the *exact* radius is required to be determined, the gage block should not be placed against the point of triangular block *D*, but against the side, and the blade of the square should rest against the angular face of the cutter instead of against the point. Then the resulting radius would be exactly equal to the thickness of the gage block. The method explained by our correspondent is, of course, convenient as a *relative* measurement, and is used as such where these fixtures are employed, but it does not give the real length of the resulting radius.—EDITOR.]

LATHE CENTERS WITH HIGH-SPEED STEEL POINTS.

In speed lathes and engine lathes used for turning small work at high rotative speeds, considerable trouble can be avoided by the use of dead centers that will stand considerable heat without softening. This can be accomplished by the use of machine steel centers with inserted high-speed steel points. The points are best made from about ½ inch round stock, having a cylindrical fit about 1 inch long. If the steel is cut 1¾ inch long, there will be ample stock for the tapered portion, and for truing up, and also for insuring a firm joint that will give no trouble from loosening or springing. For making centers of this description the following is suggested, and in my experience has proved to be a very satisfactory way. If a cutter grinder or grinding machine is available, mount the bar of round high-speed steel in the chuck, after first roughing out the cone point on a rough grinder, and grind it true and parallel. Then, swivel the head, and grind the cone point. Remove the bar from the chuck, nick it on a thin wheel, and break off the end to the length wanted. Then square up the end and bevel the corner *A*. This facilitates the pressing in place later. Finally grind a small flat on the side of the center at *B* to let the air escape when pressing it into place. If made in quantities, it is advisable to have the diameter a definite figure, so that the centers will fit a hole reamed with a standard reamer.

FOSTER F. HILLIX.

La Fayette, Ind.

OBTAINING ANGULAR MOVEMENTS WITH THE INDEXING HEAD.

In the September, 1907, issue of MACHINERY, there appeared a criticism of the article on indexing, published in the February issue, in which the critic, Mr. John Edgar, claims to have presented a clearer explanation of the relationship existing between the movements of the crank, or worm spindle, and the angular movements of the work spindle. It may be that he has done so, but I am inclined to think that the method which I shall give is preferable. Take the number of degrees passed through by the work spindle in one revolution, and divide this number by the number of teeth in the worm-wheel, or, what amounts to the same thing, the number of turns of the crank necessary to effect such a movement. The quotient will equal the number of degrees that the work spindle will move, for one revolution of the crank. In this way the relationship between the work spindle and the crank is shown as clearly as can be shown. In the criticism referred to, it was stated that a division of one degree is made by a movement of two holes in an 18-hole circle. This will be obvious from the rule which I have given, as 360 degrees divided by 40 teeth, equals 9 degrees for one turn of the crank. If the number of degrees that is to be indexed is divided by nine, the quotient will equal the number of turns that the index crank must make.

The following example is the one given by Mr. Edgar to illustrate his method of indexing: "Should it be desired to

index, say, 27 degrees, we would multiply 27 by 2 and obtain the number of holes necessary for such a movement. As 54 is a greater number than the number of holes in the circle, we would have as the movement of the index crank $54/18 = 3$ turns to make a 27 degrees division or indexing." This same result can be obtained by simply dividing 27 by 9. On the Brown & Sharpe milling machine the smallest fraction of a degree that it is possible to index exactly, by simple indexing, is 1/3 of a degree. By compound indexing, however, it it possible to index exactly 1/60 of a degree, or in other words, angles may be indexed, progressing by one minute. According to the article referred to, a division of 1/4 or 1/6 of a degree should be fine enough for even the closest jig work. Then a 1/60-degree division is evidently intended for work which must be closer than the closest jig work. But are the quarters or sixths of a degree division fine enough? Taking even the smaller fraction, the error on a 6-inch diameter is liable to be 0.0043 inch in one indexing. The error could not be more than 0.0043, as this equals the length of the arc of an angle of five minutes, the radius being 3 inches. If the work had to be indexed a few times, this error would be increased, and in that case I do not believe any one would be accused of splitting thousandths to an exaggerated degree.

Stamford, Conn. J. PRICE.

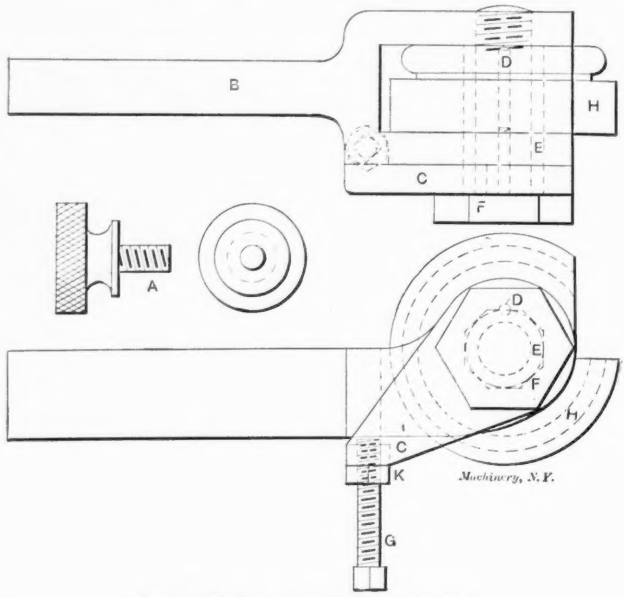
[The methods advocated by Mr. Price and Mr. Edgar are both correct, of course, and it is merely a matter of opinion which one should be preferred. To the average machinist, most likely, the fact that two holes in the 18-hole index circle represent one degree, is the one easiest to remember, and the one least likely to cause mistakes or confusion. In regard to the matter of how close angular movements may be carried on the milling machine with ordinary index heads, a certain reserve is advisable. While with compound indexing, movements of one minute are obtainable, theoretically, that does not say that with an ordinary index head such accuracy is possible when indexing, say, 30 degrees. Mr. Edgar, being a practical milling machine designer, most likely is aware of the limitations of the index head. In fact, any one who has tried to mill a side milling cutter, by using an ordinary milling machine index head, knows that when the teeth are milled on the face, and the cutter then turned so as to mill the teeth on the side, the second indexing around will not coincide exactly with the first, at least not if the indexing is not started in exactly the same relative position between cutter and index head, as when milling the teeth on the face. This indicates the practical limitations for accuracy of the index head.—EDITOR.]

IMPROVED CIRCULAR FORMED TOOL-HOLDER.

The cut herewith shows at *A* a formed knurled head screw, of which a large number had to be made, and a circular forming tool with its holder. Not having an automatic screw machine to take the size stock required, it was decided to make these screws on a lathe, using the circular tool and holder mentioned. The circular forming tool and forked tool-holder were made, and a bolt used to bind the prongs of the holder against the forming tool, no means being devised for adjusting, or preventing the tool from slipping.

The foreman of the tool-room, after experimenting in one way or another to stop the tool from slipping under the heavy strain of the cutter, thought it would be a good idea to drive a "maple wedge" in the space between the cutter and the holder. As the screws were made of cold rolled steel, and oil used for turning them, it is easily imagined to what extent the "maple wedge," after being soaked with oil, would stop the cutter from slipping. The tool was thrown aside as useless, and, after lying in its resting place for a week or more, the idea embodied in the cut occurred to me. I suggested it to the foreman, it was approved, the holder finished, and the job completed to the satisfaction of everybody.

In the cut, *H* is the formed circular tool; *B*, the forked tool-holder; *C*, a yoke with an octagon hole to fit the head of sleeve *E*, which has a keyway to fit the key *D*, the length of which is the same as the width of the inside space of the prongs of *B*. In the assembling of the tool, *D* is placed in the corresponding keyway in the hole of tool *H*, and tool and key placed in holder *B*. Then sleeve *E* is put into place, and yoke *C* is placed on the octagon head of the sleeve *E*. The hexagon clamping bolt *F* binds all together. *G* is the adjusting screw for setting the cutting edge of *H*, provided with a check nut *K*. The hole in the yoke *C*, and the head on the sleeve *E* being octagon, allows an adjustment of 45 degrees, before shifting the position of the yoke. Narrow cutters may



Improved Circular Formed Tool-holder.

be used in the same holder by placing washers or collars in the vacant space, or a gang of formed tools could also be used. This tool has been in practical service for five years, has never slipped, and has given entire satisfaction on every job used. FRANK G. STERLING.

Lowell, Mass.

PNEUMATIC DOLLY.

The small pneumatic dolly illustrated herewith, is used for "holding on" while riveting ball joint rings on the ends of locomotive dry pipes. This is not an entirely new device, but there are, doubtless, a number of shops not provided with this useful tool. The dolly is shown in detail in Fig. 1. It consists of a cylinder *A*, into which the leather-packed

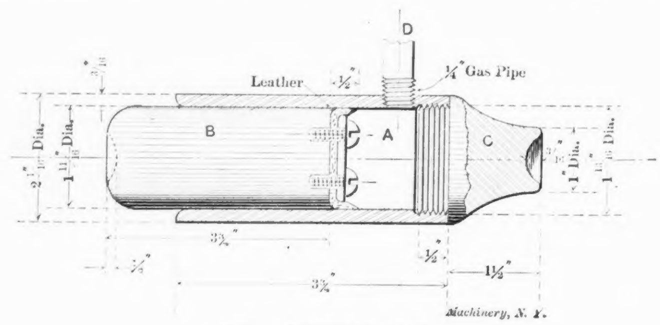


Fig. 1. Pneumatic Dolly.

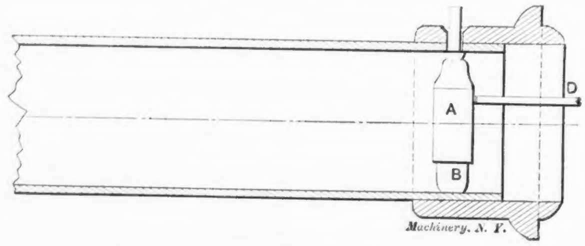


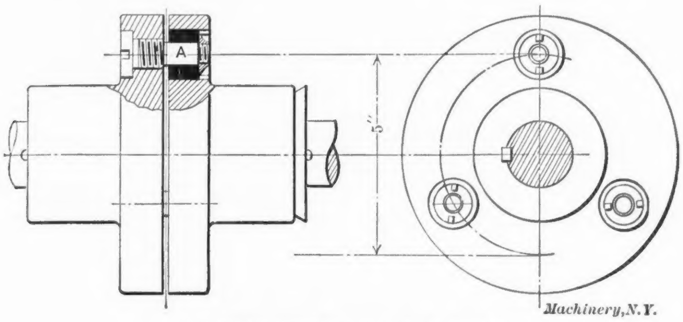
Fig. 2. The Pneumatic Dolly holding Rivet in Place.

plunger *B* is fitted. A head *C*, which is cupped to fit the head of the rivet, is held against it, as illustrated in Fig. 2, and the compressed air is admitted to the cylinder *A* by the pipe *D*. The air forces the plunger *B* against either the wall of the pipe or the head of a rivet opposite, holding the rivet to be hammered securely in place. M. H. WESTBROOK.

Port Huron, Mich.

A SIMPLE FLEXIBLE COUPLING.

At the high speeds prevailing in steam turbine practice, it it evident that careful alignment becomes necessary on direct-connected drives. A small amount of cramping and frictional retardation, due to mal-alignment, would absorb considerable power at such high speeds. One maker of these turbines of the medium and small sizes, direct connected to pumps, fan blowers and dynamos, mounts each complete outfit on a substantial bed plate, and uses a flexible coupling of the design shown in the cut. The solid black sections indicate rubber bushings, the elasticity of which permit the necessary adjustment for alignment the drive being through the bolts *A* and the bushings. With some such coupling, any springing of bed plate or differences of shaft level, due to unequal wear in bearings of



A Flexible Coupling of Simple Design.

component parts of the outfit, will not affect the friction load appreciably. The observed example is to be found on the second motion shaft of the De Laval No. 1 B engine, rated capacity 15 H. P., running at 2,400 R. P. M., and driving a Sturtevant No. 6 Monogram fan.

Figuring back to the bolt circle, with its three yielding rubber rings, the above power and speed apportions $52\frac{1}{2}$ pounds maximum pressure on each ring.

$$\frac{15 \times 33{,}000 \times 12}{5 \times \pi \times 3 \times 2{,}400} = 52.5 \text{ pounds.}$$

Apparently this pressure is below the squashing limit of the rubber, as an examination of the rubber bushings after two years' use in a warm engine room showed that they still fitted closely on the pin and in the bore of the coupling.

The groove on the hub of the coupling on the driven half, to prevent the oil from the bearing from working up and out to the flange where it might rot the rubber, should be noticed, and also the gap between the halves of the coupling for allowing adjustment of alignment, and increase of length of shafts due to temperature changes. A wedge gage or feeler tried at various places in this gap, and a straight-edge test over the edge of the flanges would quickly tell of changes in the positions of the shafts which might render a re-aligning desirable. The freedom from any projecting bolt heads seems also a good point. Other advantages may be known to the makers, but the above appear most readily to an observer.

New Britain, Conn. ROBERT S. BROWN.

[Some flexible couplings of interesting design were shown in the December, 1906, issue of MACHINERY, page 202, engineering edition, and in the June, 1907, issue, page 560, engineering edition.—EDITOR.]

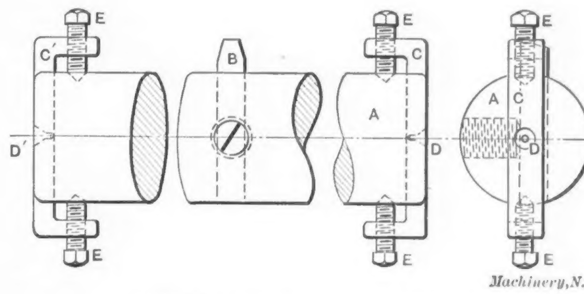
THE COLLAPSING OF HARDENED TOOLS.

In the November, 1907, issue of MACHINERY two half-tones were shown of a mandrel broken by internal stresses produced by the hardening process. An occurrence of this kind can, in most cases, be prevented, if tools are hardened in the proper manner. If we heat a piece of tool steel to a high heat, say 1,600 degrees F., and quench it in clear water until it gets entirely cold, that piece of steel is sure to become useless. Take, for instance, a mandrel or any solid tool heated and quenched in this manner. When the outside commences to cool off it shrinks, and consequently forces the mass of metal at the center together, the metal at the center still being red hot. After a while, however, the center of the piece starts to cool off also, and in cooling it will shrink. This shrinking produces tremendous strains between the outside and the center portion of the metal, because the outside portion is already cooled off, and is in a perfectly solid state, and the inside portion cannot shrink except by producing cracks in the metal. The cracks may not appear immediately, but the internal strains are in operation just the same, and will finally cause the collapse of the tool. This final collapse can be somewhat delayed by softening the mandrel on the outside, thereby relieving the strains to a certain extent, but a tool treated in this way would in most cases be too soft for the purpose for which it is intended. The proper way of hardening is, of course, to dip the heated piece of steel in cold salt water only for so long a time as to harden the outside deep enough to allow for grinding, and then place the piece in fish oil to let it cool slowly. The piece should be heated to about 1,500 degrees F. Of course we all know that if the piece is simply cooled off in water long enough to harden the outside, and then laid aside on the bench or on the floor until it cools off, it will be too soft, because the heat at the center drives the hardening from the surface. The proper method outlined, however, will insure success.

Lansing, Mich. J. F. SALLOWS.

A HANDY BORING BAR.

Although the plain boring bar with a single cutter is out of date for duplicate work, it is often useful in the tool-room, experimental room and job shop. The machinist often finds it difficult to adjust the tool of a plain boring bar, to the required accuracy, and hold it rigid. The writer has found the bar which is shown in the accompanying cut, to be very useful. It consists of a straight bar *A*, of suitable diameter and length for the work at hand, in the center of which the cutting tool *B* is held in the usual manner. Each end of the bar is milled to receive the bent pieces *C*, which are made a



Boring Bar with Convenient Adjustment.

snug sliding fit. In the pieces *C* there are center holes *D* that receive the lathe centers. By means of the adjusting screws *E*, the sliding pieces *C* can be moved across the bar, shifting the centers and moving the tool into the work as desired.

The points of the set-screws should be let into the bar as shown, for holding the pieces *C* in place while handling. By loosening one set-screw, the pieces *C* may be removed, so that the bar can be passed through the work, and the dog or driver put in place. This bar will be found of special value when cutting threads into large nuts. ARTHUR NICHOLS.

Lansing, Mich.

DRIVING THREADED STUDS IN A LATHE WITHOUT MARRING THE THREADS.

Various devices are resorted to for driving a stud held between lathe centers, or for holding a bolt or other threaded piece in a vise without marring the threads. One of the simplest and most effective methods is to saw an ordinary nut of the same diameter and pitch as the stud in two parts, cutting in a plane coinciding with the center line, or exactly as the lead-screw nut of a lathe is split. If a stock of these split nuts to fit all standard threads is kept on hand, any piece of standard size can be gripped quickly, tightly, and without damage to the threads. A nut cut through one side only is also possible to use, but has to be screwed on and off—a decided waste of time, even on short threads.

Middletown, N. Y. DONALD A. HAMPSON.

SHOP KINKS.

A DEPARTMENT OF PRACTICAL IDEAS FOR THE SHOP.

Contributions of kinks, devices and methods of doing work are solicited for this column. Write on one side of the paper only and send sketches when necessary.

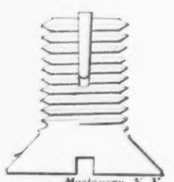
OILING CENTERS.

A very simple method for keeping the centers of a lathe properly lubricated is to tie a thoroughly oiled piece of waste around the center, so that the waste rubs against the end of the work. The pressure of the cut, of course, forces the piece of work somewhat toward one side of the center, allowing enough space to permit the oil to constantly flow into and lubricate the center hole in the work being turned. This simple method will do away with a great deal of wear on the centers, and will prevent all other unpleasant experiences which result from heated centers, not properly lubricated.

Birmingham, Ala. JOHN McLEOD.

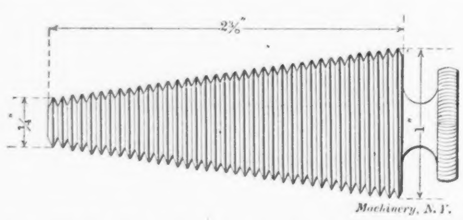
HOW TO TIGHTEN A LOOSE SCREW.

I was called upon to tighten a loose screw in a printing press. The screw was too small for the hole, and not being able to obtain one of suitable size and shape, I resorted to the following: The old screw was held between two pieces of wood in the vise, and a slot made, as shown in the sketch; the slot was opened a little, and a flat iron wedge driven in. The screw was then returned to its place, and it is now there for keeps. This method can be used also with bolts where the nut will not stay. X. Y. Z.



TOOL FOR CALIPERING TAPER TAPPED HOLES.

The tool shown in the cut is used for calipering taper tapped holes in boilers when fitting studs. It is a simple, though very useful and economical tool, and it will doubtless be appreciated by those having much work of this kind to do.

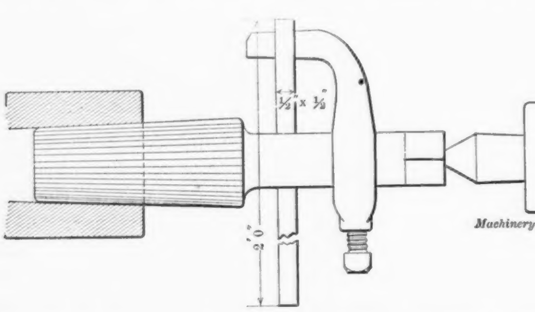


The hole in which the stud is to be fitted is calipered by filling the threads of the plug with chalk, and then screwing the plug in the hole. When the plug is removed the chalk will show exactly the largest diameter of the hole.

Brighton, Mass. F. RATTEK.

REAMING A TAPER SLEEVE.

I have seen many mechanics fuss around for hours when reaming a taper sleeve, and, finally, go to the boss for help. When doing work of this kind, the reamer is supported and guided by the dead center, and kept from turning by a lathe

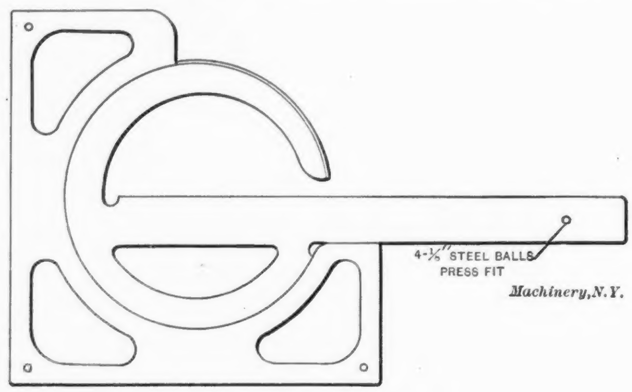


dog which is fastened to it. When held in this manner, the reamer tends to feed into the work and tear itself from the dead center. This can be prevented by the following method: After roughing the work to the desired size, prepare to ream the hole as usual. Then procure a light pine stick about ½ inch square by 2 feet long, and place this stick in the position shown in the cut. Hold the stick with the left hand, and pull slightly toward the dead center. Start the lathe slowly, and with the right hand feed the reamer into the work. This stick will enable one to feel the cutting of the reamer, and the feed can be judged accordingly. If the reamer should bind, the stick will strike the lathe and break, and the work and reamer will be free to turn. If a little care is given to the feed and speed, this method of holding a reamer will be found satisfactory. W. H. ADDIS.

Decatur, Ill.

RAISING DRAFTSMAN'S PROTRACTOR ABOVE THE SURFACE OF THE DRAWING.

Anyone who uses a Brown & Sharpe draftsman's protractor is familiar with the way it soils the drawing, when put to any extended use, in spite of the utmost care on the draftsman's part. The cut herewith shows a scheme which, I believe, is original with Mr. Geo. L. Merrill, of this city, for



raising the surface of the protractor slightly from the drawing. Four steel balls are made a press fit in the device at about the positions shown, the balls being about ⅛ inch diameter, projecting an equal amount on both sides, thus giving the same results no matter which side of the instrument is up. This addition to the instrument in no way affects its accuracy, and, as it bears only on four points, it rubs less dirt into the drawing than an ordinary triangle.

Detroit, Mich. M. R. KAVANAGH.

MULTIPLE THREAD CUTTING ON THE ENGINE LATHE.

Where there are many multiple-thread screws of *coarse* pitches and leads to be cut it is a valuable kink for a machine department foreman to know that engine lathes having lead-screws even multiples in pitch of the screws to be cut, are the most convenient to use, other things being equal. When given a multiple-thread screw to cut, most mechanics divide the change gear on the stud into as many parts as there are threads, say two for double, three for triple, and so on. This applies when the stud and spindle run at the same rate. After cutting one thread, the parallel thread or threads are located by slipping the change gear and setting for the new positions by the chalk marks on the change gear. This method takes time and is liable to cause mistakes. Quite often it happens that a change gear cannot be used having a number of teeth that is an exact multiple of the multiple thread. For example, a triple thread would require the change gear to have such number of teeth as will be exactly divisible by 3. If such gears are not available, then there is trouble.

If, instead of sending all such thread cutting to one lathe, we distribute it according to the character of the lead-screws, there will be considerable gain of time and less chance for spoiled work. Suppose, for example, that we have a quadruple-thread screw, 1 inch lead, to cut. If a lathe having a 4-threads-per-inch lead-screw can be used, the setting of the tool for each of the parallel threads will simply be a matter of opening the split nut and moving the lathe carriage one thread of the lead-screw and then closing the nut. If the screw to be cut was a triple thread of 1 inch lead, a 6-pitch lead-screw could be used, moving the carriage two threads of the lead-screw, and so on. R. H. MITCHELL.

New Castle, Ind.

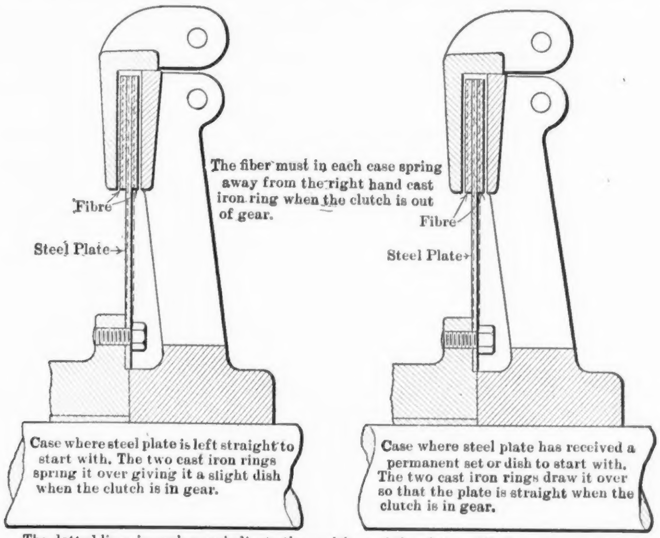
HOW AND WHY.

A DEPARTMENT INTENDED TO CONTAIN CORRECT ANSWERS TO PRACTICAL QUESTIONS OF GENERAL INTEREST.

Give all details and name and address. The latter are for our own convenience and will not be published.

DESIGN OF STEEL PLATE FRICTION CLUTCH.

D. G. M.—In the case of a steel plate friction clutch in which the steel plate with its fiber covering springs away from the cast iron face when it is out of engagement, which is the better practice: to give the plate a slight dish so that when the clutch is engaged it will straighten the plate into a perfectly flat disk, or to leave it straight and to spring it into a slightly dished shape when the clutch is engaged? The cuts Figs. 1 and 2 will illustrate the meaning clearly.



The dotted lines in each case indicate the positions of the plates with their fibre coverings when the clutch is in gear. Clutches are shown out of gear. *Machinery, N.Y.*

Figs. 1 and 2.

A.—Our preference would be to leave the plate flat and to spring it into the required shape and position by the action of closing the clutch. This will avoid the difficulty of making the plate run true after dishing it. The available clearance being small, it is essential that it run very true, and this condition is best obtained with a flat plate. The practical action otherwise should be about the same. The question is submitted to our readers for suggestion and comment.

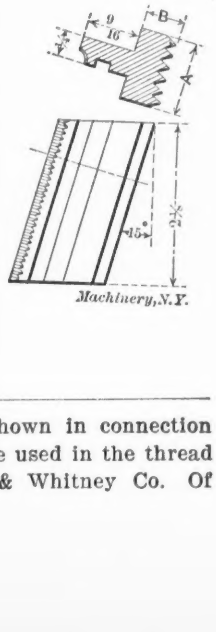
THREADING CHASERS.

Tool-maker.—Is there any rule regarding the number of threads with which ordinary threading chasers should be provided on their face? I have seen chasers with as many as 20 threads or teeth on their face, but usually they seem to be provided with only six or seven.

A.—We infer that when speaking of threading chasers our correspondent refers to such as are held in standard threading tool holders. The chaser which is shown in connection with the accompanying table, is of the type used in the thread tool holder manufactured by the Pratt & Whitney Co. Of course, the data given herewith apply to any class of threading chaser of a similar type. There is no rule governing the number of teeth with which a chaser should be provided, but it is customary to increase the number of teeth for finer pitches as compared with the coarser ones. The accompanying table gives all dimensions necessary for these tools. The dimensions given conform to accepted practice.

DIMENSIONS OF THREADING CHASERS.

No. of Threads per inch.	A	B	No. of Teeth in Chaser.	No. of Threads per inch.	A	B	No. of Teeth in Chaser.
3	1.333	3/4	4	12	0.667	5/16	8
3¼	1.231	3/4	4	13	0.615	5/16	8
3½	1.143	3/4	4	14	0.571	1/4	8
4	1.000	5/8	4	16	0.500	1/4	8
4½	1.111	5/8	5	18	0.500	1/4	9
5	1.000	1/2	5	20	0.450	1/4	9
5½	0.909	1/2	5	22	0.409	3/16	9
6	0.833	1/2	5	24	0.375	3/16	9
7	0.714	3/8	5	26	0.385	3/16	10
8	0.750	3/8	6	28	0.357	3/16	10
9	0.667	3/8	6	30	0.333	3/16	10
10	0.700	3/8	7	32	0.312	1/8	10
11	0.636	3/8	7	36	0.278	1/8	10
11½	0.696	3/8	8	48	0.250	1/8	12



FIGURING LATHE CHANGE GEARS.

Apprentice. Please state in a simple manner the way of figuring change gears for a lathe. I have seen some text books on the subject, but they all seem to make the subject so complicated that I do not fully understand the methods explained.

A.—While the principles and rules governing the calculation of change gears are very simple, they, of course, presuppose some fundamental knowledge of the use of common fractions. If such knowledge is at hand, the subject of figuring change gears, if once thoroughly understood, can hardly ever be forgotten. It should be impressed upon the minds of all apprentices that the subject in itself is extremely simple, and that the difficulty usually presents itself because the matter is not approached in a logical manner, and is usually grasped by the memory rather than by the intellect. Before answering the question in regard to any rules for figuring change gears, let us therefore analyze the subject. The lead-screw B of the lathe (see Fig. 1) must be recognized as our first factor, and the spindle as the second. If the lead-screw has six threads per inch, then, if the lead-screw makes six revolutions, the carriage travels one inch, and the thread-cutting tool travels one inch along the piece to be threaded. If the spindle makes the same number of revolutions in a given time as the lead-screw, it is clear the tool will cut six threads per inch. In such a case the gear D on the spindle stud J, and gear E on the lead-screw are alike. If the spindle makes twice the number of revolutions of the lead-screw, the spindle revolves twelve times while the tool moves one inch, and consequently twelve threads per inch will be cut. But in order to make the spindle revolve twice as fast as the lead-screw, it is necessary that a gear be put on the spindle stud of only half the diameter of the gear on the lead-screw, so that when the lead-screw revolves once the spindle stud gear makes two revolutions.

Simple Gearing.

Suppose we wish to cut nine threads per inch with a lead-screw of six threads per inch, as referred to above. Then the six threads of the lead-screw correspond to nine threads on the piece to be threaded, which is the same as to say that six revolutions of the lead-screw correspond to nine revolutions of the spindle; or in other words, one revolution of the lead-screw corresponds to 1½ of the spindle. From this it is evident that the gear on the lead-screw must make only one revolution while the spindle stud gear makes 1½. Thus, if the lead-screw gear has, for instance, 36 teeth, the gear on the spindle stud should have only 24; the smaller gear, of course, revolving faster than the larger. If we express what has been previously said in a formula we have:

$$\frac{\text{threads per inch of lead-screw}}{\text{threads per inch to be cut}} = \frac{\text{teeth in gear on spindle stud}}{\text{teeth in gear on lead-screw}}$$

Applying this to the case above, we have:

$$\frac{6}{9} = \frac{24}{36}$$

The values 24 and 36 are obtained by multiplying 6 and 9, respectively by 4. By multiplying both the numerator and the denominator by the same number, we do not change the proportion. As a general rule we may then say that the change gears necessary to cut a certain number of threads per inch are found by placing the number of threads in the lead-screw in the numerator, the number of threads to be cut in the denominator, and then multiply numerator as well as denominator by the *same* number, by trial, until two gears are obtained, the number of teeth of which are both to be found in the set of gears accompanying the lathe. The gear with the number of teeth designated by the new numerator is to be placed on the spindle stud (at J, Fig. 1), and the

gear with the number of teeth corresponding to the denominator on the lead-screw *B*.

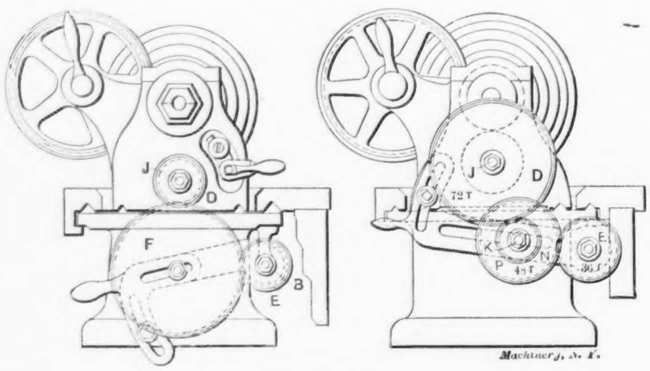
A few examples of this will more clearly define the rule. Suppose the number of teeth of the change gears of a lathe are 24, 28, 32, 36, and so forth, increasing by 4 teeth up to 100. Assume that the lead-screw is provided with 6 threads per inch, and that 10 threads per inch are to be cut. Then,

$$\frac{6}{10} = \frac{6 \times 4}{10 \times 4} = \frac{24}{40}$$

By multiplying both numerator and denominator by 4, we obtain two available gears with 24 and 40 teeth, respectively. The 24-tooth gear goes on the spindle stud, and the 40-tooth gear on the lead-screw. Assuming the same lathe and gears, let us find the gears for cutting $11\frac{1}{2}$ threads per inch, this being the standard number of threads for certain sizes of pipe thread. Then,

$$\frac{6}{11\frac{1}{2}} = \frac{6 \times 8}{11\frac{1}{2} \times 8} = \frac{48}{92}$$

It will be found that multiplying with any other number than eight would, in this case, not have given us gears with such number of teeth as we have in our set with this lathe. Until getting accustomed to figuring of this kind, we can, of course,



Figs. 1 and 2.

only by trial find out the correct number by which to multiply numerator and denominator. The number of teeth in the *intermediate* gear *F*, Fig. 1, which meshes with both the spindle stud gear and the lead-screw gear is of no consequence.

Lathes with Reduction Gears in Head-stock.

In some lathes, however, there is a reduction gearing in the head-stock of the lathe, so that if equal gears are placed on the lead-screw and the spindle stud, the spindle does not make the same number of revolutions as the lead screw, but a greater number. Usually in such lathes the ratio of the gearing in the head-stock is 2 to 1, so that with equal gears the spindle makes two revolutions to one of the lead-screw. This is particularly common in lathes intended for cutting fine pitches or, in general, in small lathes. In figuring the gears this must, of course, be taken into consideration. As the spindle makes twice as many revolutions as the lead-screw with equal gears, if the ratio of the gears be 2 to 1, that means that if the head-stock gearing were eliminated, and the lead-screw instead had twice the number of threads per inch as it has, with equal gears the spindle would still revolve the same as before for each inch of travel along the piece to be threaded. In other words, the gearing in the head-stock may be *disregarded, if the number of threads of the lead-screw is multiplied by the ratio of this gearing.* Suppose, for instance, that in a lathe the lead-screw has eight threads per inch, that the lathe is geared in the head-stock with a ratio of 2 to 1, and that 20 threads are to be cut. Then

$$\frac{2 \times 8}{20} = \frac{16}{20} = \frac{16 \times 4}{20 \times 4} = \frac{64}{80},$$

which two last values signify the number of teeth in the gears to use.

Sometimes the ratio of the gearing in the head-stock cannot be determined by counting the teeth in the gears because the gears are so placed that they cannot be plainly seen. In such a case, equal gears are placed on the lead-screw and the spindle stud, and a thread cut on a piece in the lathe. The number of threads per inch of this piece should be used for the numerator in our calculation instead of the actual number of threads of the lead-screw. The ratio of the gearing in the head-stock is equal to the ratio between the number of threads cut on the piece in the lathe and the actual number of threads per inch of the lead screw.

Compound Gearing.

The cases with only two gears in a train referred to are termed simple gearing. Sometimes it is not possible to obtain the correct ratio excepting by introducing two more gears in the train, which is termed compound gearing. This class of gearing is shown in Fig. 2. The rules for figuring compound gearing are exactly the same as for simple gearing excepting that we must divide both our numerator and denominator into two factors, each two of which are multiplied with the same number in order to obtain the change gears.

Suppose a lathe has a lead-screw with six threads per inch, that the number of the teeth in the gears available are 30, 35, 40 and so forth, increasing by 5 up to 100. Assume that it is desired to cut 24 threads per inch. We have then,

$$\frac{6}{24} = \text{ratio},$$

By dividing up the numerator and denominator in factors, and multiplying *each pair of factors by the same number*, we find the gears:

$$\frac{6}{24} = \frac{2 \times 3}{4 \times 6} = \frac{(2 \times 20) \times (3 \times 10)}{(4 \times 20) \times (6 \times 10)} = \frac{40 \times 30}{80 \times 60}$$

The last four numbers indicate the gears which should be used. The upper two, 40 and 30, are driving gears, the lower two, with 80 and 60 teeth, are driven gears. Driving gears are, of course, the gear *D*, Fig. 2, on the spindle stud, and the gear *P* on the intermediate stud *K*, meshing with the lead-screw gear. Driven gears are the lead-screw gear, *E*, and the gear *N* on the intermediate stud meshing with the spindle stud gear. It makes no difference which of the driving gears is placed on the spindle stud, or which of the driven is placed on the lead-screw.

Suppose, for a final example that we wish to cut $1\frac{3}{4}$ threads per inch on a lathe with a lead-screw having six threads per inch, and that the gears run from 24 and up to 100 teeth, increasing by 4. Proceeding as before, we have

$$\frac{6}{1\frac{3}{4}} = \frac{2 \times 3}{1 \times 1\frac{3}{4}} = \frac{(2 \times 36) \times (3 \times 16)}{(1 \times 36) \times (1\frac{3}{4} \times 16)} = \frac{72 \times 48}{36 \times 28}.$$

This is the case directly illustrated in Fig. 2. The gear with 72 teeth is placed on the spindle stud *J*, the one with 48 on the intermediate stud *K*, meshing with the lead-screw gear. These two gears (72- and 48-teeth) are the *driving* gears. The gears with 36 and 28 teeth are placed on the lead-screw, and on the intermediate stud as shown, and are the *driven* gears.

* * *

Interesting experiments on the shrinkage of wood due to the loss of moisture have recently been completed by the Forest Service at its timber testing station at Yale University. These experiments show that green wood does not shrink at all in drying until the amount of moisture in it has been reduced to about one-third of the dry weight of the wood. From this point on to the absolutely dry condition, the shrinkage in the area of cross-section of the wood is directly proportional to the amount of moisture removed. The shrinkage of wood in a direction parallel to the grain is very small; so small in comparison with the shrinkage at right angles to the grain, that in computing the total shrinkage in volume, the longitudinal shrinkage may be neglected entirely. The volumetric shrinkage varies with different woods, being about 26 per cent of the dry volume for the species of eucalyptus known as blue gum, and only about 7 per cent for red cedar. For hickory, the shrinkage is about 20 per cent of the dry volume, and for long leaf pine about 15 per cent. In the usual air-dry condition, from 12 to 15 per cent of moisture still remains in the wood, so that the shrinkage from the green condition to the air-dry condition is only a trifle over half of that from the green to the absolutely dry state.

NEW MACHINERY AND TOOLS.

A MONTHLY RECORD OF APPLIANCES FOR THE MACHINE SHOP.

CHANDLER CLUTCH-DRIVEN PLANER.

The Chandler Planer Co. of Ayer, Mass., has developed a clutch-driven planer of remarkable interest. It is shown in the accompanying half-tone. As may be seen, the driving mechanism, which is the principal feature of the improvement, has been applied to a planer of the frog and switch type, a machine which is subjected to about the hardest service that is ever imposed on a planer, thus giving a first-rate opportunity to try out the value of the new mechanism.

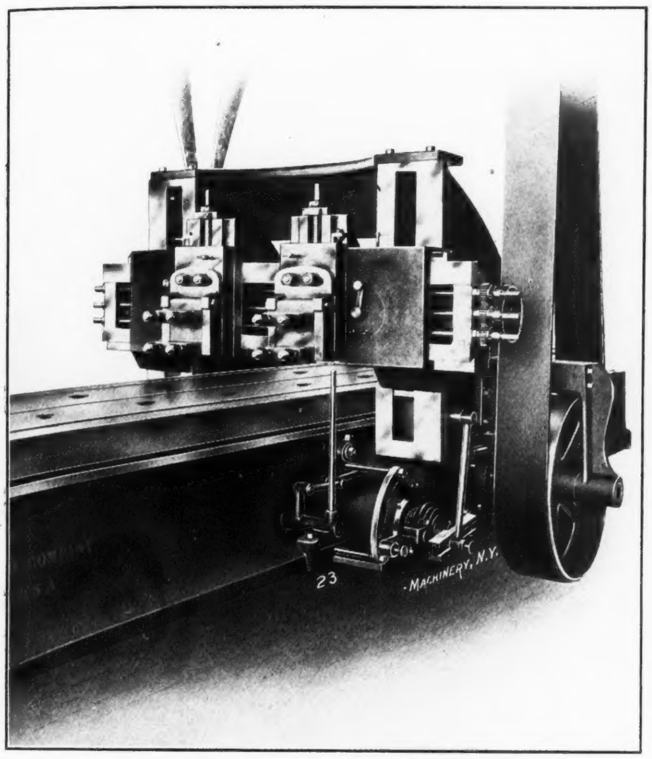


Fig. 1. The Chandler Clutch-driven Planer.

The problem of driving a planer becomes serious as the size of the machine and the severity of the duty is increased. Wide belts are required to transmit the tremendous power required for taking heavy chips at the high speeds possible with modern steels, and for reversing the table and work at these high speeds. The width of the belt can be reduced to some extent by increasing its velocity, but the fast running of the driving pulleys thus made necessary tends to defeat the object for which it was used, by increasing the momentum of the rotating parts and making the reversal much more difficult than it was before. This matter is very serious, since it has been found that the greater part of the power required for reversal (which is the greater part of the power required for driving the machine) is expended in overcoming the inertia of the rotating pulleys and gears.

There being a limit, then, to the speed at which belts can be run, and the increased power necessitating the widening of the belts, the problem of shifting them from the tight to the loose pulley and *vice versa* becomes exceedingly serious in heavy service. To obviate this difficulty, the use of clutches has been proposed for reversing in the place of shifting belts. Two forms of these clutches have been used more or less successfully—the electromagnetic and the pneumatically operated. The first has been found in practice to offer difficulties in the way of giving sufficient holding power, getting rid of residual magnetism and avoiding wear on contact faces. The pneumatic clutch has been tried on a considerably larger scale, and has been considerably used. Among its disadvantages is the fact that it requires an air-compressing plant—something which every shop does not possess, and which may have to be put in separately as an adjunct to the machine. Furthermore, it is not positive and invariable in its action, but may be changed, adjusted and regulated to suit the whim of the operator who, very likely, is better acquainted with the operation of the planer than with that of compressed-air machinery. Finally it acts with a given force which may or may not be enough to transmit the power desired. If not, the machine has to be stopped and the mechanism adjusted, this being necessary occasionally as the parts wear and conditions of lubrication, humidity of the atmosphere, etc., change.

In the machine under consideration, the clutch is operated mechanically. The problem to be solved is that of making a clutch and operating mechanism which will be strong enough to drive the heaviest cut the tools used will stand, and reverse the heaviest work at the highest speed the planer will ever be called on to work at—this to be done without shock, and without wear of the contact areas due to slipping. Of course there must be some slip to stop the planer table gently, and start it back on its return trip without violent shock. The problem is to allow just enough slip to effect this without giving so much that the contact surfaces rub on each other for a considerable period of time, thus wearing each other away.

The mechanism used in this machine effects all this in a remarkably ingenious fashion. The amount which the contact surfaces slip on each other before gripping is regulated by the design, and this amount is maintained under all conditions of lubrication and atmospheric moisture. Furthermore, it is as impossible for the clutch to slip as it is for a positive clutch. Something would have to break first. The way in which this is done may be briefly described by saying that the planer dogs do not directly operate the main clutches which connect the machine with the forward or reverse driving pulleys. The shifting of the reversing mechanism first throws a supplementary clutch into action with the driving shaft. The difference in motion between this supplementary clutch and the main driving clutch, by means of suitable mechanism, operates to engage the latter by power until the two parts are moving together. If slip occurs at any time, the

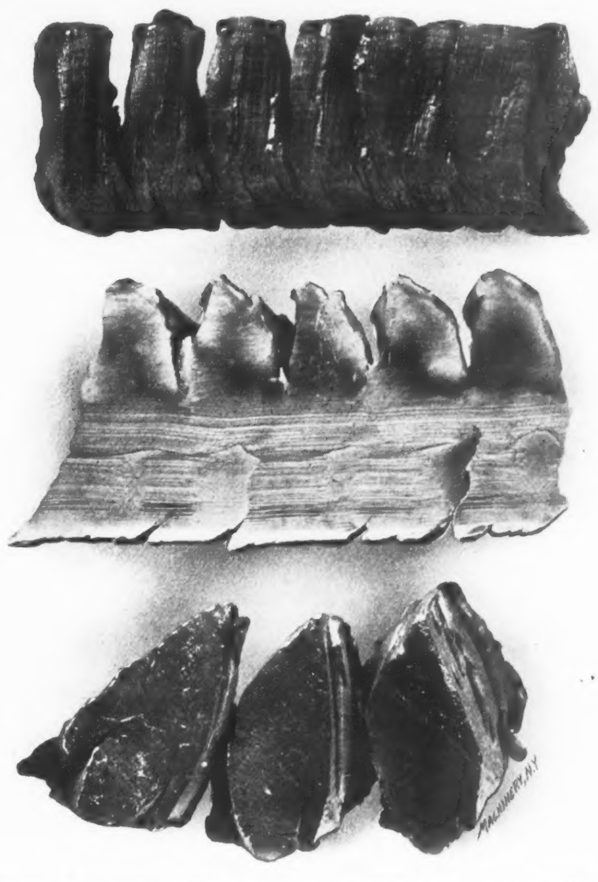


Fig. 2. An Example of Chips taken on the Clutch-driven Planer—Full Size.

slipping only serves to tighten the connection the firmer. In reversing, the throwing of the secondary clutch operates first to release the main clutch previously in operation, and then to engage the other one.

The writer witnessed a test of the frog and switch planer to which this mechanism has been applied. It was interesting to a high degree and at times really exciting. The test was intended to show that the planer had been developed to a point where it exceeded the capabilities of the steels with which it was used. In other words, in any work which it may be desired to perform on the planer, the cutting speed, and the depth and width of the cut will be limited by the tool, rather than by the machine. In demonstration of this, tools of high-speed steel of approved form and temper and unusual size were repeatedly tested to destruction. One way of doing this was by placing a 40-point carbon steel forged slab on the planer bed, and taking a series of wide and deep chips from it, one after the other, stopping each chip a little before the conclusion of the previous one. After five or six of these cuts had been made, the tool was again started in and the planer table allowed to continue its full stroke. The tool, thus catching up with one stopping point after another, took a gradually heavier and heavier chip until it finally had more than it could bear and broke short off at the shank. One of those broken measured 2½ x 3 inches at the broken section.

The chips thus produced, of which we show samples in Fig. 2, were interesting not only on account of their unusual size, but as well in showing the way in which chips are produced. This was fully explained in Mr. Taylor's paper "On the Art of Cutting Metals," the action being illustrated in Figs. 1, 2 and 3 in the February, 1907, issue of MACHINERY. As there explained, the chip is separated from the main body of the metal by alternate compressing and shearing, the different shearing planes separating the chip into sections more or less completely, depending on the heaviness of the cut taken. These separate sections are plainly shown. In many cases during the test the cuts were so heavy that these sections separated entirely, there being no continuous chip, but instead, a series of chunks of metal which flew in a stream from the tool, striking against a nearby pile of castings with a rattle like the discharge of a rapid-fire gun. The operation, in fact, seemed distinctly dangerous, and suggested the use of an armor-plate shield for the man operating the machine. Of course, these are extreme conditions, and were only undertaken to show the utmost capabilities of the machine.

During the tests the cutting speed of the table was 20 feet per minute, with a return of 4 to 1. The double cutting belt is 10 inches wide and runs at a speed of 1,860 feet per minute. The reversing belt is 5 inches wide, running at 3,900 feet per minute. The planer removed the side of the head up to the web on a pair of extra hard 70-pound relay switch points, cutting back 9 feet, in less than six minutes. At the same cutting speed it removed a chip ¾ inch deep and ¾ inch feed in a 40-point carbon steel forging. In doing this work, the tool required about 6 to 7 horse-power on the stroke and about 20 horse-power for reversing.

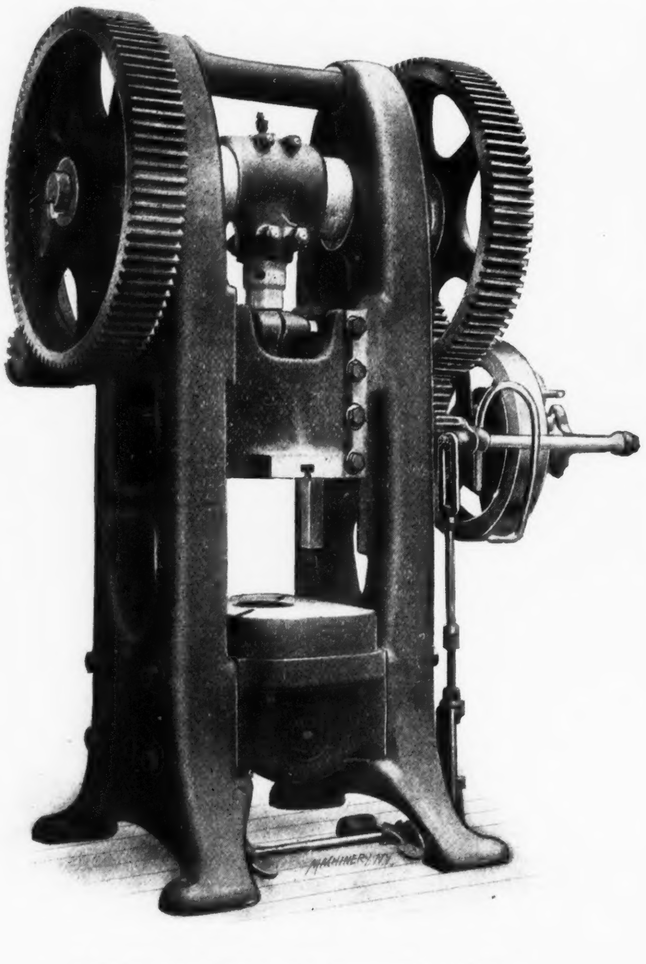
This planer has been in operation now for about three months, and has so far shown no tendency toward deterioration in the wearing parts of the clutch, which is of the multiple disk type. The disks which form its bearing surfaces are of cast iron permanently submerged in a bath of oil. After running for this length of time, the disks have been disassembled and examined, and the cast iron bearing surfaces have been found to be glazed, showing practically no signs of wear. This is, of course, the best condition for wearing in the case of cast iron surfaces in contact. It is stated that when the clutch has been disassembled while still tightly engaged under service conditions, and with the oil drained from the clutch, a layer of oil has been found between each of the separate disks, tending to show that they do not actually come in contact at all, there always being a film of lubricant between them.

The inventor of this machine is Mr. D. L. Chandler, superintendent of the firm and inventor of the three-belt planer as regularly built by this firm, and described in the July, 1904, and December, 1907, issues of MACHINERY.

FERRACUTE SINGLE-CRANK PRESS.

The accompanying illustration shows a massive single-crank press just built by the Ferracute Machine Co., of Bridgeton, N. J. In common with other machines of this kind, the press has a long stroke which adapts it for drawing cold from heavy sheet steel, deep drawn seamless shells, automobile hubs, cups for ball-bearings, and similar work. It can be built with shorter stroke for heavy blanking, trimming, shearing and embossing. A somewhat larger press of this series recently built by the Ferracute Machine Co. for the United States government is being used in the manufacture of cartridge shells for cannon.

The castings which compose the frame of this machine are exceedingly heavy, with neatly rounded corners which add considerably to the appearance of the press. Other striking features are the heavily trussed bed and the wide-faced twin gears on each side of the press. These gears are cut from the solid. Having two gears on the crank-shaft instead of one, tends to relieve the torsional stress of the shaft and affords an even pressure, besides dividing the load between



A Single-crank Press built by Ferracute Machine Co.

them. The back-shaft, with its two pinions which engage the large gears, is a single steel forging, made strong enough to obviate any torsional effect.

The main shaft is forged from high carbon steel. The clutch and fly-wheel are mounted on a stud which is rigidly attached to the frame, thereby preventing the disturbance of alignment which is apt to occur when such supports are detached and bolted to the floor. The press is equipped with a "multi-disk" friction clutch of a new and effective design. It is of the automatic stop type, but can be quickly adjusted for stopping the ram by hand at any point of its descent or ascent; or, if desired, it can be set for continuous running. Positive knockouts are provided when desired, the press being designed so they can be readily attached.

The distance between columns is 28 inches. The stroke is 3 inches, but can be made, if ordered, of any length up to 17 inches. The height from the bed to the ram when raised is 19½ inches, the adjustment of the ram being 6 inches. The fly-wheel is 35 inches in diameter with a 6-inch face,

and weighs 750 pounds. The press occupies a floor space of 8 feet 7 inches from right to left and 5 feet 5 inches front to back, and is 10 feet 7 inches high. It weighs 25,000 pounds, and exerts a pressure of 200 tons.

"LEKTRO" PORTABLE UNIVERSAL GRINDER.

This little portable tool is designed to perform, in combination with standard machine tools, all the functions of the various kinds of grinding machines in common use. It may be used on the lathe, milling machine or planer, for cylindrical, internal and surface grinding, and for sharpening of all sorts of tools, such as milling cutters, reamers, counterbores, circular forming tools, etc. Cutters may be ground while in place on the milling machine when so desired.

This attachment consists primarily, as may be seen in Fig. 1, of an electric motor mounted on an angle plate or knee, on which it may be raised or lowered by means of the elevating screw shown. The motor is especially designed for this work, the end motion of the spindle being prevented by an accurately made thrust bearing. The motor shaft is of tool steel, hardened, ground and lapped to size. The armature disks are ground on separate arbors before being assembled on the finished armature shaft, and are balanced after winding to insure smooth running. The commutator is balanced separately, and the whole armature tested for balance when assembled. The bearings are in phosphor bronze bushings, tapered and split, and adjusted by threaded collars. Oil cups are provided on both sides of this bearing, so as to allow the motor to run in an inverted position. The field magnet of the motor is so compactly arranged as to allow work to be brought to within 1 9/16 inch of the axis of the shaft on the front or working side of the tool. The whole motor can be swung on the knee which carries it to any desired angle, and can be completely turned around in the opposite direction if desired.

The half-tone engravings shown herewith are nearly self-explanatory, and serve to illustrate a few of the varied uses to which the tool can be adapted. In Fig. 1 the live center of a lathe is being trued up, the compound rest being set at the proper angle for that purpose. In Fig. 2 is shown an example of cylindrical grinding on the lathe, this being done on dead centers, a special drive being arranged for the purpose. The spindle cone is disconnected from the spindle, and a bushing is clamped to the live center, carrying a pulley

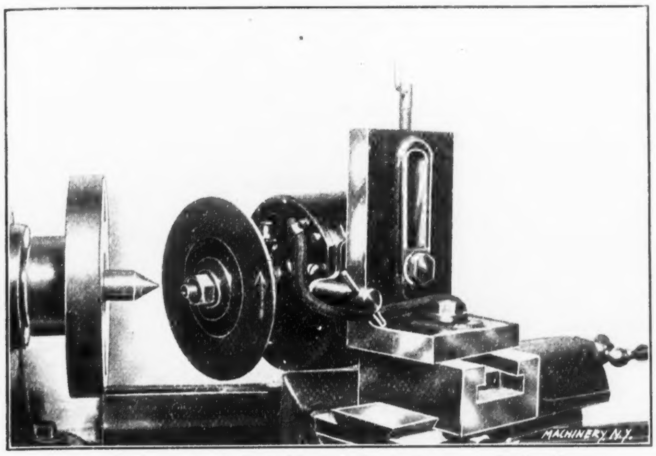


Fig. 1. The "Lektro" Portable Grinder truing up a Lathe Center.

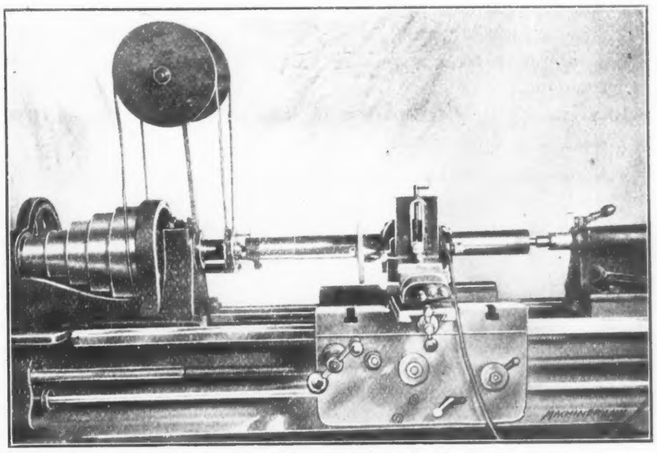


Fig. 2. Cylindrical Grinding in the Lathe.

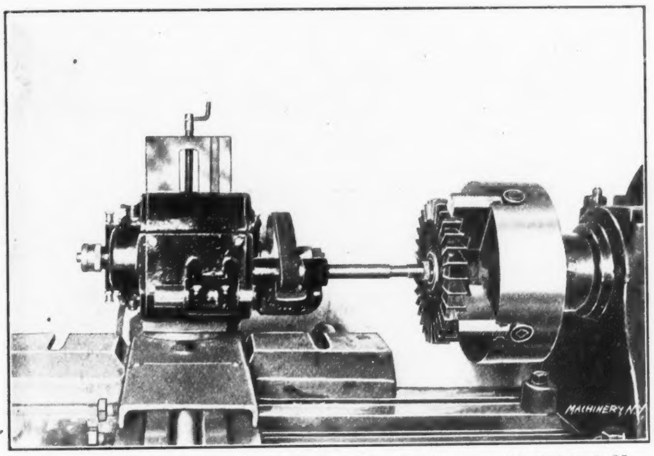


Fig. 3. Internal Grinding Attachment. The Spindle runs 14,000 R. P. M.

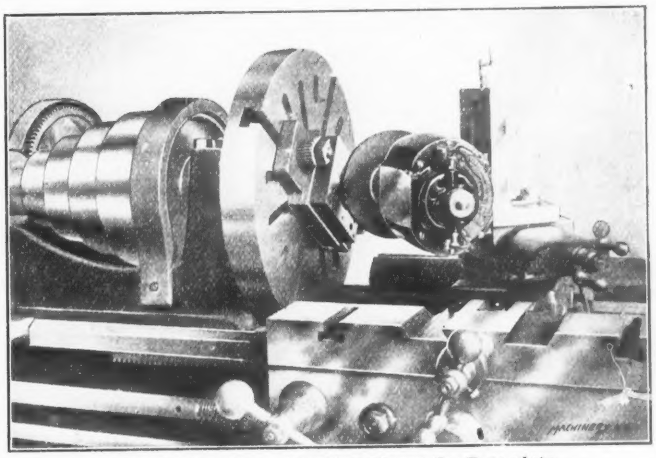


Fig. 4. Grinding a Blanking Die on the Face-plate.

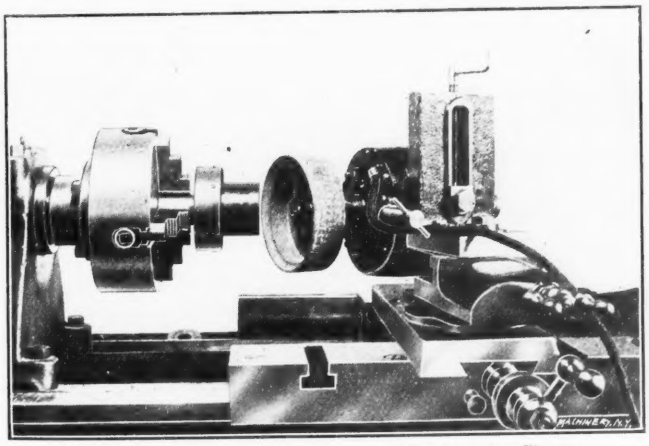


Fig. 5. Grinding a Punch with a Cup Wheel to give Shear.

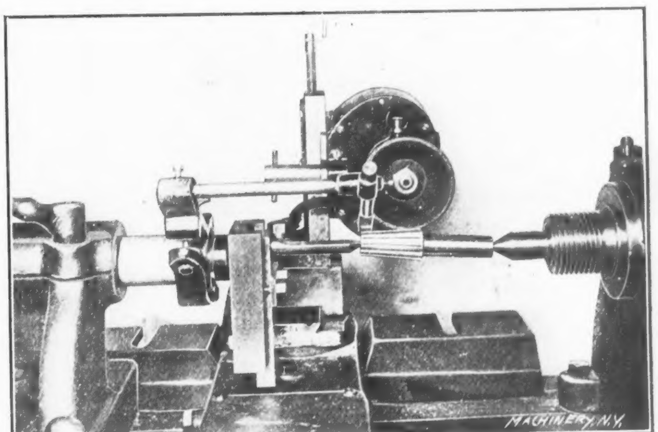


Fig. 6. Sharpening a Taper Reamer in the Lathe using Adjustable Tail Center.

loosely revolving on it. This is belted over the quarter turn counter-shaft shown, to the large step of the spindle cone. A spring-supported dead center is provided similar to that used in grinding machine practice.

In Fig. 3 the tool is being used with an internal attachment which is supplied with it. This attachment consists of a bracket fastened to the front face of the motor, and carrying a holder having bearings for the small internal spindle, which is driven by a short belt from a pulley mounted on the motor shaft in place of the usual emery wheel. In this operation, the hole in a hardened cutter is being ground to size.

Figs. 4 and 5 show examples of the grinding of press tools. Fig. 4 is practically a surface grinding operation. The die which is being sharpened is clamped to the face-plate, and a facing cut taken over it by the grinding wheel. Fig. 5 shows a cup wheel being used for grinding a round punch. The angle at which it is set makes it possible to grind the cutting edge with any desired amount of shear.

clearance desired for the cutting edge of the face. The cut is taken by working the longitudinal feed of the slide. A hook stop is used, and the belt is preferably run off from the cone pulley, which is operated by hand to index the work against the stop. In Fig. 9 the outside cutting edges are being sharpened. In this case the stop is mounted on the table, and the cross-feed of the saddle is used for the grinding operation. Fig. 10 is unusually interesting in showing the ability of the device to perform grinding operations of a delicate character. Here we have an end mill of very small diameter, having the teeth on its end sharpened. The operation is essentially the same as that of Fig. 8, the added difficulties being due to the small size of the work.

Of course these ten figures merely give an idea of a few of the operations for which the device may be used. Its use as a surface grinder in connection with the planer will be readily suggested, while its application to internal and external grinding of all kinds, and the sharpening of cutters, ream-

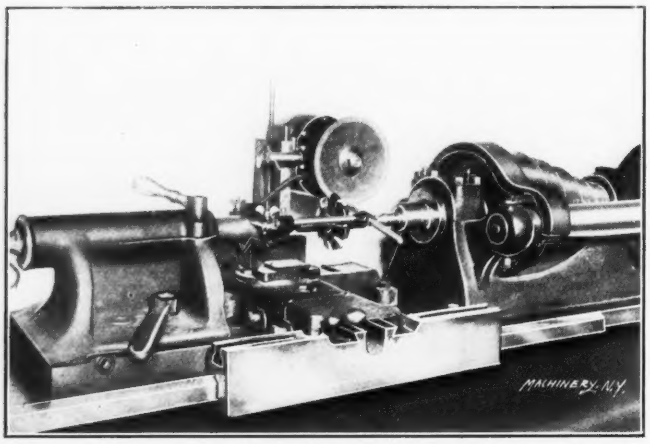


Fig. 7. Sharpening a Gear Cutter in the Lathe with Tooth Rest clamped to the Tail-stock Spindle.

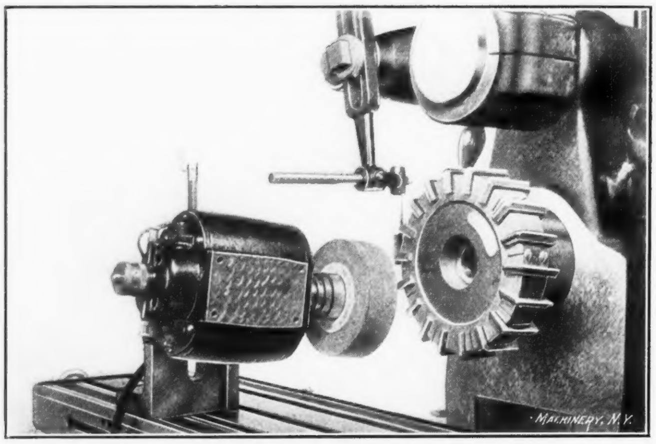


Fig. 8. Sharpening the Face of a Heavy Face Mill in place on the Milling Machine.

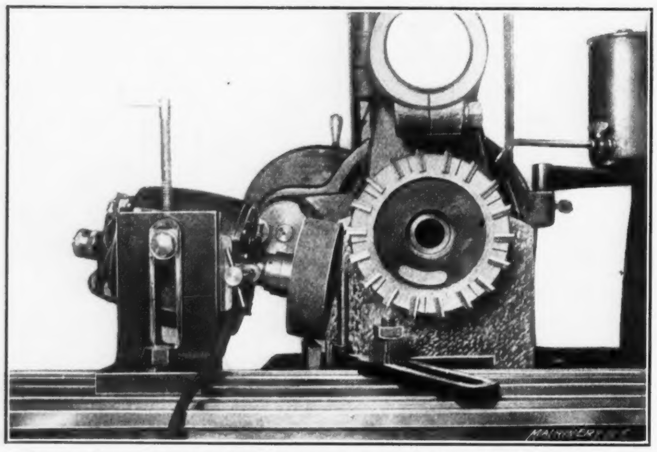


Fig. 9. Sharpening the Periphery of a Face Mill, the Blades of which are set at a Slight Angle.

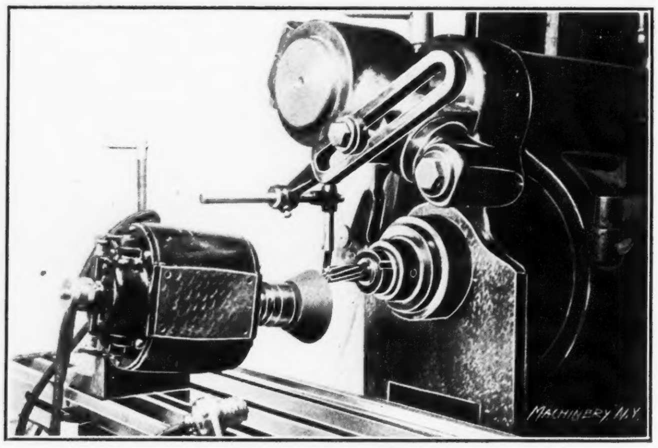


Fig. 10. Sharpening the End Teeth of a Small End Mill in place in the Milling Machine Spindle.

The next two figures show the machine used for sharpening small tools in the lathe in combination with attachments provided for the purpose. In Fig. 6 an adjustable center is used in place of the regular solid center. It may be raised or lowered to suit the taper it is desired to grind. A taper reamer is shown in place between the centers. A tooth-rest or stop is provided, held by clamping straps to the tail-stock spindle. This may be adjusted in various ways to suit different kinds of cutter and reamer work. In Fig. 7 a formed gear cutter is being ground, in this case on an arbor between the regular centers. The tooth-rest is again shown in operation. These cuts are, of course, taken by actuating the carriage by the usual hand-wheel.

Figs. 8, 9 and 10 show another class of cutter grinding for which this tool is particularly adapted—namely, the sharpening of cutters in place on the milling machine, under such circumstances as to insure their running true, with every tooth doing its duty. In this case the tooth-rest is supported in any convenient way to suit the case in question. In Fig. 8 it is supported from the overhanging arm. The large face mill is being sharpened in this case. A cup wheel is used on the grinder, which is set at the proper angle to give the

ers, taps, mills, etc., is well nigh universal. This machine is manufactured by the "Lektro" Mfg. Co., 44 Walnut St., Cincinnati, O.

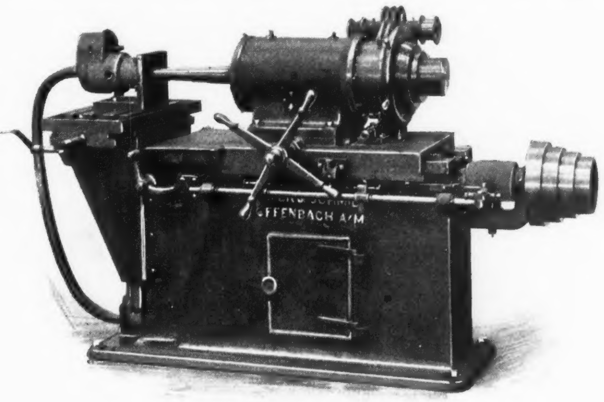
AUTOMATIC CYLINDER GRINDER.

The illustration shows an automatic cylinder grinding machine, manufactured by Mayer & Schmidt, Offenbach-on-Main, Germany, which was exhibited at the Olympia Exposition, London. The photograph was received too late to permit it to be included in Mr. James Vose's notes on the exposition, published last month.

This machine is designed to work completely automatically in all its movements, and an attainable accuracy within 0.00025 inch is claimed. The grinding head, which, judging from the cut is unusually rigid, is mounted on a carriage which moves back and forth during the grinding operation. The cylinder to be ground is mounted on a table which has adjustment both sideways and in a vertical direction, which provision makes it easy to get the axis of the cylinder in line with the axis of the grinding head. During the grinding operation, however, the cylinder is stationary.

The wheel-carrying arbor or spindle of the grinding head

has, besides its rotary motion, also a motion around the axis of the grinding head, the grinding spindle being eccentric in relation to this. The amount of this eccentricity is easily adjustable. The machine is also provided with a scale by means of which an automatic stop can be engaged when the desired diameter in the cylinder has been reached, at which time a bell rings, and the grinding spindle recedes automatically from the work. All feeds are automatic, and a cone feed pulley is provided so that a great variation of the forward and



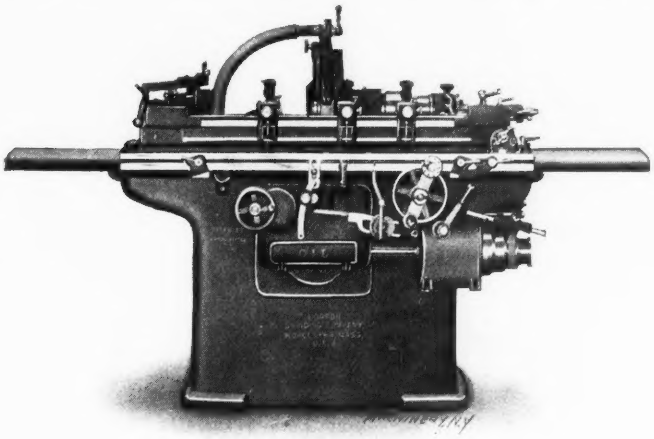
Automatic Cylinder Grinder.

reverse carriage feeds is obtainable. The grinding spindle is driven through a movable counter-shaft which follows the eccentric motion of the spindle. With the exception of the grinding spindle itself, which, of course, must be long enough to reach through the work, over-hanging parts have been avoided so far as possible, and this, together with the perfect rest during the operation of the cylinder which is to be ground, makes a high degree of accuracy possible. These machines are built in sizes to grind from 4 inches to 12 inches inside diameter, by 8 inches up to 44 inches in length.

SMALL NORTON GRINDING MACHINE.

We show herewith a new size of the well-known line of grinding machines made by the Norton Grinding Co. of Worcester, Mass. This machine is the smallest of the line, as built at the present time. It is made to swing the work up to 6 inches in diameter and 32 inches long, though its ordinary working range is for work from ½ to 2½ inches in diameter, the steady-rests being of suitable design for shafts of this character.

The well-known Norton system of universal steady-rests is used. Three of these appliances are furnished with the



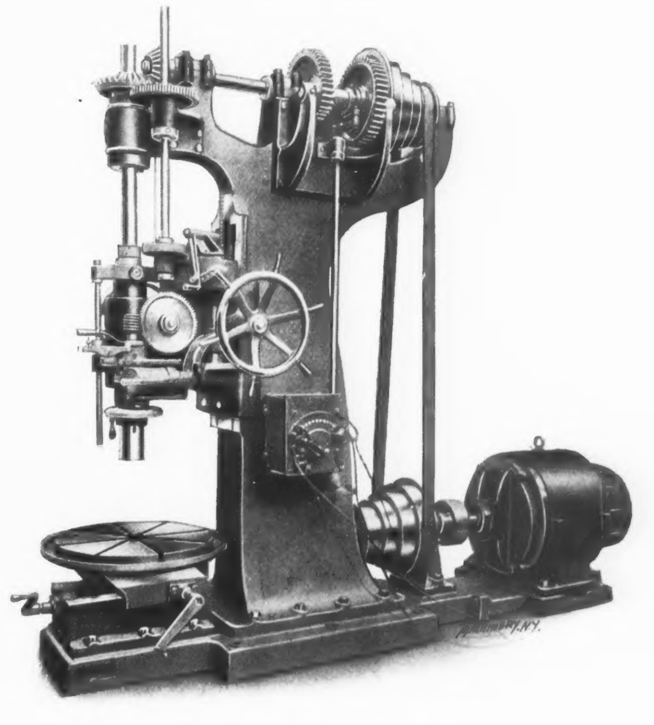
A New Addition to the Line of Norton Grinders.

machine, supplied with a set of work shoes for one diameter of work. Other sizes can be made from blue-prints furnished with the machine. These shoes are designed with a view to quick change from one size work to another. They all furnish a good rigid support for long work, within the range of sizes given. Provision is made for grinding tapers up to 2 inches per foot. A center grinding attachment is furnished with this machine, arranged to grind center points round, true, and to an accurate 60-degree angle. If a different angle is required, a special attachment can be made.

The over-head works furnished with the machine is of improved design, with shafts made of the best material, ground on centers. The hangers have large, self-oiling bearings. The weight of the machine with the over-head works, is about 3,600 pounds. The tool is arranged for belt drive only.

AN ENGLISH HIGH-SPEED DRILL PRESS.

The half-tone illustrates a drilling and boring machine built by J. Butler & Co., Halifax, England. This style of machine is built in two sizes, taking 30 inches and 36 inches diameter, respectively. Among the points included in the construction may be mentioned a fixed and rigid work table with adjustments in each direction by quick pitch screws, and through which the weight is taken solidly on the base plate; a strong drill head-stock fitted to the vertical column and balanced for quick and easy adjustment to the height of the work; and a very substantial spindle, which is supported in a steel sleeve with gun metal lining and ball thrust bearings at top and bottom. The sleeve has a slow hand traverse and a quick hand adjustment, each instantly disengaged or engaged, and governed by an automatic trip motion. The positive feed is



An English High-speed Drill Press.

driven by spur gears from the spindle, and is variable either by a three-speed feed box, or quadrant plate with change gears as desired. Further points are the powerful drive by a pair of steel bevel gears—the one on the spindle having a long sleeve in the upper bearing, and driving by two long keys. The driving shaft has capped bearings with gun metal bushings, and is driven by the speed cones directly, or through the double gear. As will be seen, the machine is self-contained, the counter-shaft being fixed on the base of the machine with starting lever close at hand, the illustration showing the machine driven by electric motor, direct. In the case of belt driving from the main shaft, the motor is replaced by fast and loose pulleys with belt shifting device. The machines may be made to drive by speed cones only, or with the double gear arrangement, enabling heavy boring to be done. In this case the speeds are arranged to give as gradual a progression as possible. Some details of the machines are appended.

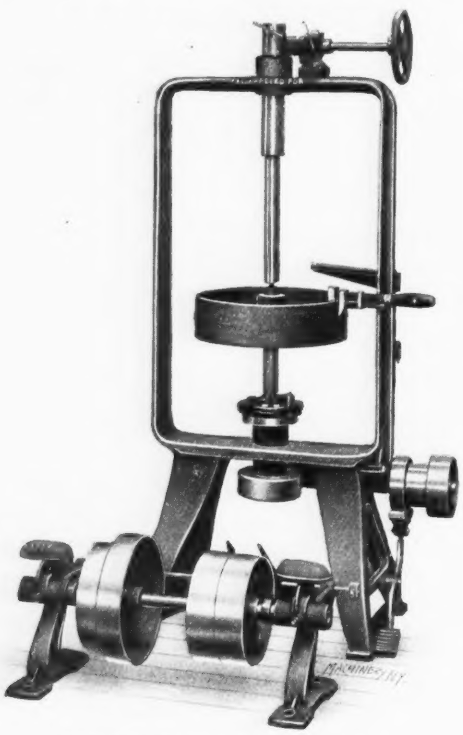
The 30-inch machine admits from spindle end to top of table 24 inches, and to base plate 42 inches. The 36-inch machine admits 28 inches and 46 inches. The diameters of the spindles at the driving end are 2 and 2¾ inches respectively; the diameters in the sleeves 2¼ and 3 inches, and the diameters of the sleeves 4½ and 6 inches, the bores of the

spindles being Nos. 4 and 5 Morse taper. The travels of the spindles are 12 and 16 inches, and the vertical range of the drill head on face of uprights 12 inches. The table diameters are 24 and 30 inches. In the 30-inch machine, the number of speeds on the cone is three, and in the 36-inch machine, four, the widths being 3 and 3½ inches, and the largest diameters 18 and 20 inches. The ungeared 30-inch machine will drill from ½ to 1½ inch holes at 600 to 270 revolutions per minute, and the 36-inch size will drill from ¾ to 3 inch holes at 600 to 183 revolutions per minute. When geared, 3 and 6 inch holes represent the duty recommended, the speeds varying from 600 to 83 revolutions in one machine, and from 600 to 39 in the other. The machines weigh 7,280 and 9,850 pounds respectively. Though the duty given is that which high-class high-speed drills will perform, the machines themselves are powerful and speedy enough to ruin any twist drill within their capacity. One of the machines was used for demonstrating the capacity of high-speed drills at the Olympia Exhibition, and in one test a ¾-inch drill was fed at the rate of 25 inches per minute, and a 2½-inch drill at 2½ inches per minute. J. V.

CURRIER REAMING MACHINE.

The machine illustrated in the accompanying half-tone is of an entirely new design, and is intended for performing reaming operations by mechanical means, at a great saving of time and with greater accuracy than is possible to be had when reaming by hand. The obstacles to be overcome in a reaming machine are mainly the difficulty of starting the reamer in the hole without roughing the end of the hole, which also includes the difficulty of starting the reamer straight, and the troubles met with from the digging in of the teeth of the reamer, which cannot as easily be provided for in a reaming machine as in hand reaming. Both of these difficulties have, however, been taken care of in the present machine, the work being guided until the reamer has fully entered the hole, and an equalizing drive, referred to later, provided to insure uniformity and ease of cutting action.

In the upper and lower ends of the frame 2½-inch holes are bored in perfect alignment. The lower end is fitted with a spindle, which is driven by a worm-gear drive. This spindle



A Machine for Reaming by Power.

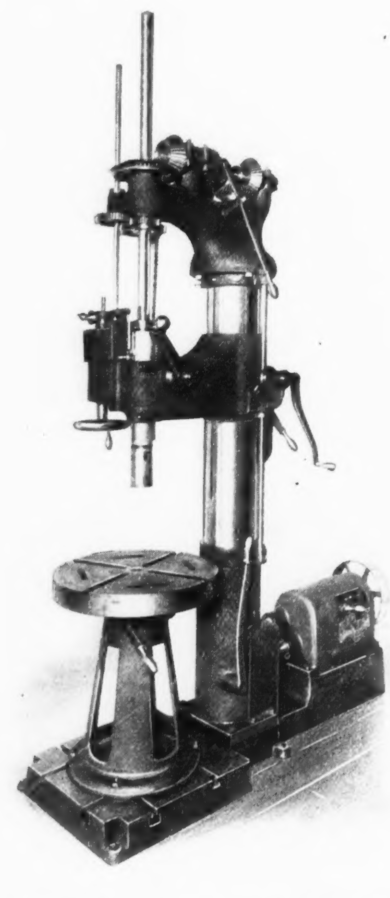
is in turn fitted with a center and an equalizing carrier for driving the reamer. At the upper end of the frame a sleeve is fitted, which is operated vertically by a rack and pinion. In this sleeve a bar is held, upon which the work to be reamed is placed until the reamer is put on the centers, one end of this bar containing what we might call the tail center. A brake attached to the lower side of the right-hand leg makes it possible to stop the machine instantly by foot power.

The work to be reamed is first chucked in the usual manner and reamed, leaving from 0.004 to 0.005 inch. The work is then placed on the guide bar, and a swinging arm, shown on the right-hand upright, brought forward to prevent the work from dropping while the reamer is placed on the centers. When the reamer is in place, the machine is started, the swinging arm is thrown to one side, and the gravity of the work feeds it slowly down over the reamer. To prevent the work from turning, it may be held by a wrench, as shown in the cut, or by any similar suitable device. It will be noticed also that the reamer is fluted through the shank; this provision is made in order to permit the chips to fall through easily. The simplicity of the machine permits it to be run by a great deal cheaper help than could be employed for hand reaming.

The machine will take in work up to 24 inches diameter. It is built by J. E. Snyder & Son, Worcester, Mass., and is furnished with counter-shaft, six carrier wrenches, and one guide bar of any size from 1 inch to 2½ inches. The speed of the counter-shaft is 400 R. P. M., and the weight of the machine is 430 pounds.

DRILL PRESS OF NOVEL DESIGN.

The cut herewith shows a 24-inch upright drill press, built by the National Machine Tool Co., Cincinnati, Ohio, termed by the makers their 1907 design. As will be noticed, it is



The National Machine Tool Co.'s 24-inch Drill Press.

designed differently from the ordinary type of upright drills, being without back support or top braces. But none the less, for a machine of its size, this drill is particularly rigid.

The head of the drill rotates with the sleeve about the column, an arrangement which will be found advantageous for heavy work. The vertical adjustment for the head is by means of a crank, bevel gearing and screw, as plainly shown in the cut. Twelve changes of speed, arranged for drills from ½ inch to 2½ inches, are provided for. One-half of these speeds are obtainable by the speed box, which provides for six changes of speed, it being possible to make changes while the machine is running. This number of speeds is multi-

plied by two through the back gears. The latter are also designed to be thrown in while the machine is running.

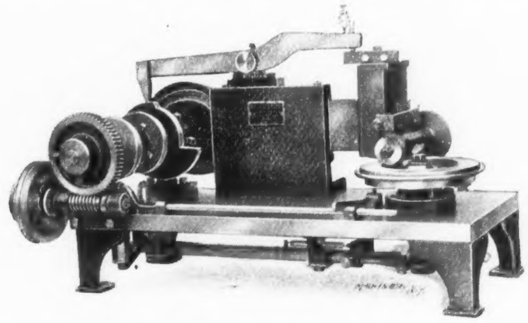
There are four feed changes, the feed change mechanism being located directly on the front of the head. The quick return and approach lever also engages and disengages the feed, so that but one hand needs to be used for these operations. The tapping attachment is placed in the base of the column, behind the back gears, which insures greater power. The starting, stopping, and reversing lever is right under the hand of the operator, as shown in front of the column in the cut. The style of table used insures alignment at all times. It is rigid and stiff, and has a revolving top. The table, as is seen from the cut, is easily removable, and drilling on the base can be performed without much trouble experienced in first removing the table.

The general dimensions of the machine are as follows: Total height of drill, 7 feet 5 inches; diameter of column, 8 inches; Morse taper in spindle, No. 4; maximum distance between the spindle and base, 48 inches; between spindle and table, 23 inches; diameter of table, 22 inches; working surface on base, 22 x 22½ inches; spindle speeds, from 36 to 267 R. P. M.; feeds per revolution of spindle, 0.008, 0.012, 0.016 and 0.020 inch; speed of driving pulley, 360 R. P. M. The floor space required is 22½ x 55 inches, and the total weight of the drill is 2,100 pounds.

AUTOMATIC SLOTTING MACHINE.

The machine shown in the accompanying engraving is a modification of the automatic pinion cutter built by the Standard Manufacturing Co. of Bridgeport, Conn. It is shown engaged in the operation of sawing slots in the base and edge of a ring, which is part of a knitting machine. The mechanism is cam-operated throughout.

In this particular case, the cut taken is a peculiar one. In the figure, the cutter is set in a position to start a cut. It travels toward the outer edge of the ring, then travels down past the edge of the saw, making a perfect right-angle cut in the work. The slide carrying the cutter is forced down by a mechanism which, in the standard pinion cutting machine, is used for withdrawing the cutter on the return stroke during the indexing, and bringing it down again to cutting depth.



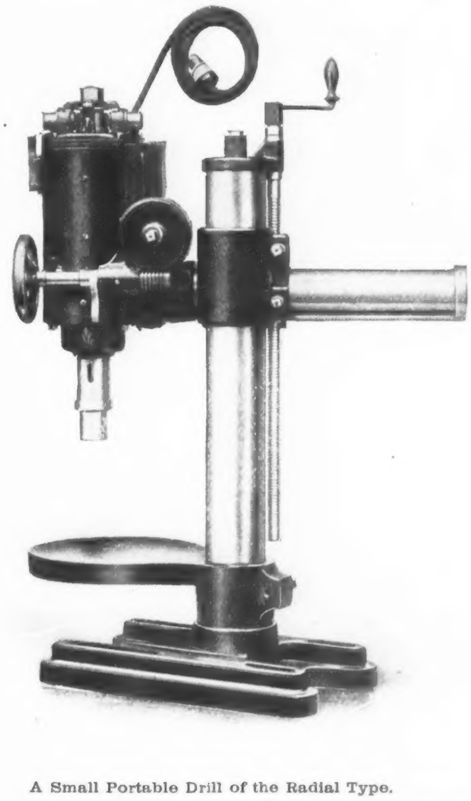
A Machine for automatically Sawing Radial Slots.

The index mechanism is so arranged as to be adjustable to the desired number of cuts, by altering the number of teeth that the ratchet will take. A positive lock is provided, which holds the index disk firmly while the saw is doing its work. Provision is also made for holding this disk while the work is being removed from the table or tightened down upon it. This is done in such a way that no strain is thrown on the indexing mechanism, so that all possibility of throwing the machine out of alignment is avoided. This machine is also sold arranged for slotting screw heads of all sizes and styles, a hopper feed being used in conjunction with it.

PORTABLE ELECTRIC RADIAL DRILL.

In the accompanying cut a new portable tool made by the Hisey-Wolf Machine Co., Cincinnati, O., is illustrated. This tool is termed by its makers the Hisey portable electric Scotch radial drill, of 2-inch capacity. These drills are also made in two smaller sizes of ⅞-inch and 1¼-inch capacity, respectively. These drills are self-contained, and are portable in the full sense of the word, being of a comparatively small size. The range of work that can be done on these machines is, within their capacity, as complete as that of the large size types of stationary radial drills, but on account of their small weight, the machines can be taken to any part of the shop, or outside of the works, without trouble, one man being able to easily handle a machine of this size. These drills are electrically driven, and are made for either direct or alternating current. The motor has two speeds, controlled by a thumb lever at the lower end. The driving power is obtained from an ordinary lamp socket. This permits the drill to be used anywhere within the machine shop.

The 2-inch capacity drill, which is the one illustrated, has a 12-inch feed through the hand-wheel shown at the left, and



A Small Portable Drill of the Radial Type.

is arranged for quick return by hand. It can drill at a radius of 24 inches, at any place in the horizontal plane, and at any angle. The hand-wheel and the worm box are provided with a swivel adjustment so that the tool may be used in corners and where space is limited. The horizontal column, as well as the vertical arm, is made of hollow steel tubing, and the column is fitted with a screw to raise and lower the motor and drill head, as shown. The bracket on the horizontal column is also fitted with a pinion meshing with a rack on the vertical arm, which provides the radial adjustment. These attachments permit the operator to adjust the drill to his work very quickly. The table is adjustable and detachable. As may be judged from the illustration and description, the machine is thus provided with possibilities for quick adjustments in all respects, and will therefore, undoubtedly, be found a very useful tool wherever a small portable drill is required.

POWER HACK-SAW GIVING RELIEF ON THE RETURN STROKE.

The Racine Gas Engine Co. of Racine, Wis., is building a power hack-saw which has mechanism designed to raise the blade of the saw on the return stroke. This operates without interfering with the feeding mechanism. The builders state that it gives surprising results in the way of length of life of the blades, rapidity of cutting, and accuracy of the cut. Fig. 1 shows the general appearance of the machine; the line cut, Fig. 2, will serve to explain the action quite clearly.

A is the guide on which the saw frame travels, while *B* is a weight adjustable on the bar on which it is mounted for the degree of feed desired. The saw frame is reciprocated by a crank in the usual fashion. The crank is mounted directly on the driving pulley shaft, the pulley being large enough in diameter so that no gearing is required. Fastened to guide *A*,

which is pivoted around the driving shaft, is a sector *C*, curved to the arc of a circle with the same center as the guide, and having ratchet teeth on its concave surface. On the driving shaft is a cam *D*, half of the periphery of which has a somewhat greater diameter than the other half. This operates a roll on lever *E* in such a way that the latter is raised at the beginning of the return stroke and lowered at the beginning of the cutting stroke. The outer end of lever *E* carries a dog *F*, engaging the teeth in sector *C*. When this lever with the dog is raised as the cutting stroke is completed, the sector is raised with it, and the saw returns free

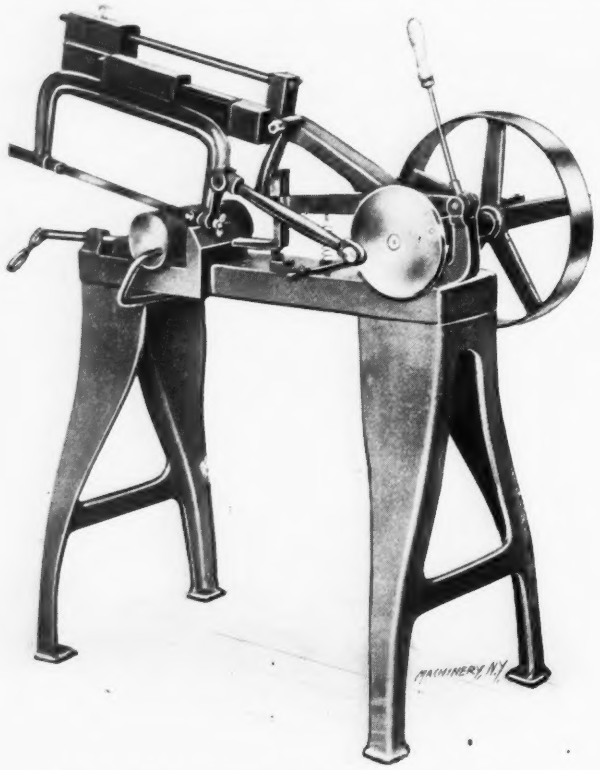


Fig. 1. Racine Automatic Relieving Power Hack-saw.

of the work, not dragging as is the usual case. At the end of the backward stroke, as the saw is about to return, cam *D* allows lever *E* to drop again, and the saw is again lowered in contact with the work. The long bent tail of the dog *F*, when it is lowered, rests on the head of screw *G*, and is by it thrown out of contact with sector *C*, leaving the saw thus entirely free to be acted on by the feeding weight *B*.

Lever *H*, operating a clutch, is connected with the slide in which screw *G* is held in such a way that the latter is

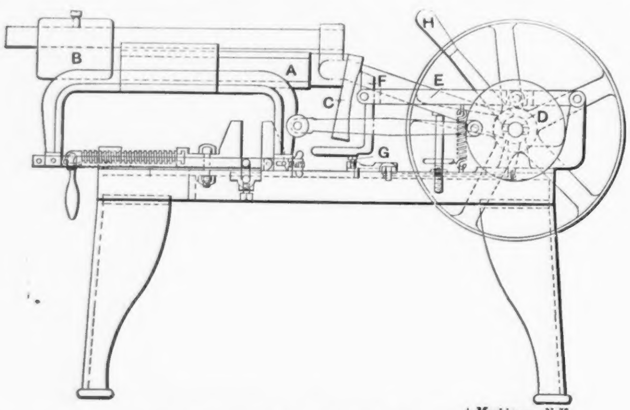


Fig. 2. The Mechanism of the Racine Power Hack-saw.

moved out of contact with the tail of the dog when the machine is stopped. This allows the saw to be raised and held in any position when it is not in operation, making it convenient for inserting work or taking measurements, etc. A stop is provided for depth of cut in cases where it is not desired to cut clear through the piece. The clutch is connected with an automatic stop arrangement which throws out the power connection when the work has been cut through. Provision is made for using either 10-inch or 14-inch blades, depending on the size of the work, thus allowing the full length of the blade to be used in any case.

This machine was the result of the necessities of its builders, who had large quantities of crucible machinery steel to saw. They experienced great difficulty in doing this, owing to the fact that the saw would become so dull before the cut was completed that it would run out of true from ¼ to ½ inch, and sometimes would not go clear through without breaking the blade, thus wasting time and material, besides the extra facing required in the lathe. After the lifting device

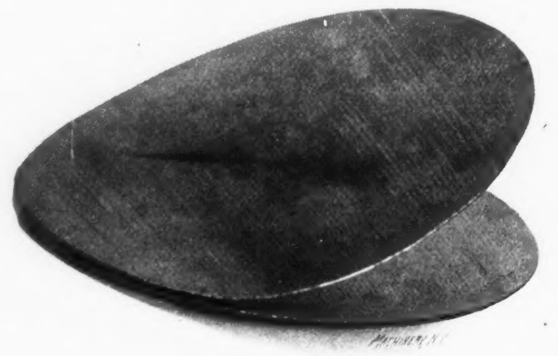


Fig. 3. A Sample of the Fine Work of which the Machine is Capable.

was applied, there was no trouble on the same work in taking six cuts with one blade, these being so nearly square that the stock was cut to within 1/16 inch of finished length. As an example of the excellent work of which the saw is capable, we show in Fig. 3 a reproduction of a photograph of a section of crucible steel in which a cut has been taken separating a piece of only about 0.040 inch thickness, there being a variation of less than 0.010 inch over the whole area of the cut.

SHEPARD ELECTRIC HOIST.

The electric hoist shown in two forms in the two accompanying half-tones has been designed after careful reference to operative requirements of such mechanisms, and it is believed by its builder (the General Pneumatic Tool Co., Montour Falls, N. Y.) to fill these requirements very nearly. Perhaps the appliance can best be described by taking up these requirements in turn, and explaining how the design, in each case, has been made to conform to them.

One of the features of this hoist, plainly in evidence in Fig. 1, is its compactness. This has not been considered as a primary virtue, and nothing in the way of accessibility and strength has been sacrificed to obtain it. It will be seen, however, that the mechanism has been enclosed in an unusually small casing or framework. The ruggedness of construction necessary for a machine to be used largely by unskilled workmen has been carefully looked out for, also, all its load sustaining parts being of unusual strength. With the simple design chosen, this has been found possible without exceeding the dimensions or weight of other hoists of equal rated capacity.

Interchangeability has been carefully provided for. Not only is the hoist built on a plan which makes it possible to replace repair parts without fitting, but parts made for these hoists will interchange with corresponding parts in traveling cranes built by the same makers as well. This plan results in greater economy of manufacture also, and makes it possible to furnish at current prices a much better hoist than would otherwise be possible. The hoists themselves may be considered as interchangeable parts to be applied indiscriminately as simple hoists, or as parts of electric cranes and other apparatus of a similar nature. As all these machines of the same capacity will interchange perfectly, very few spare parts will suffice for a large and varied equipment. This interchangeability extends to the pneumatic form of the hoist as well, which is identical with the electric form in many of the mechanical features of its design, and so may be interchanged with it in any given case.

Careful attention has been given to the matter of accessibility, no other feature of the design having been allowed to interfere with this prime requirement. This has been carried out to the extent that every important group of parts

is accessible without disturbing unrelated mechanism. For instance, the armature can be removed without disturbing any part of the electrical connections except the armature leads, and requiring the removal of no part of the mechanism except the front housing of the motor. The monkey wrench and screw driver are the only tools needed to take the machine apart or reassemble it.

In the design of this tool the severe use to which it is likely to be put, and the lack of skill in the workmen who are expected to use it have required that adjustments be entirely done away with. All wearing surfaces are made to take permanently their proper position, and are designed with sufficient bearing area so that any disturbances due to natural wear will not affect their perfect operation. Any attendant of average ability or less will have no difficulty in removing or replacing parts successfully and safely, since no delicate adjustments are involved.

Constant and thorough lubrication is necessary in any machine of this kind, no matter how well designed and made it may be. Provision is taken in these hoists to insure the access of oil to all running parts, even when proper attention to this important matter is not given by the operator or attendant. Every part of the hoist will be properly oiled when three conspicuous oil reservoirs are filled. The supply of lubricant contained in each of these three reservoirs is sufficient to protect the machine from harm, even if it is not replenished for a long period.

An important improvement consists in providing the motor with a variable speed electric control. It has been usually considered that this is not necessary with series-wound motors of small size, but experience has led the builders to believe that the sudden turning on of full current strength into a

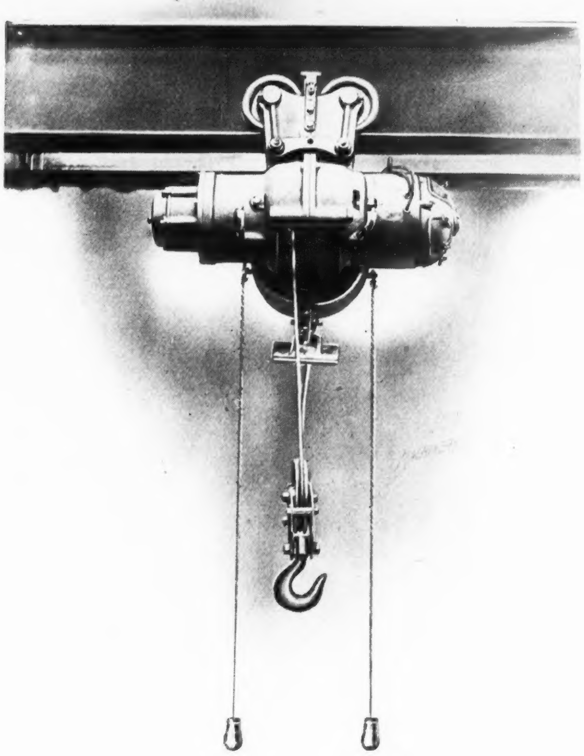


Fig. 1. The Shepard Electric Hoist.

hoist motor results in unnecessary strain on the shafts, gears and hoisting rope, and when long continued, results in serious deterioration in the motor as well. On opening the circuit under full voltage and current strength, there is also the serious damage to the controller contacts to reckon with. The controller used with these hoists provides an unusual number of contact steps—14 on the smallest size, and 18 in those of larger capacity. It is proven in practice that this improvement results in a great decrease in the amount of care necessary to maintain the electrical features of the hoist in good condition.

As intimated in the paragraph devoted to the matter of interchangeability, these devices may be applied in a number of different ways, using the proper structure or holding mechanism for the work in hand. In Fig. 1 is shown the simplest form of the hoist, arranged to be supported by a single I-beam, and to be traversed by hand. The controller in this case is mounted on the farther side of the hoisting drum, and is operated from the floor by pull ropes. An automatic stop is provided to prevent over-winding. In Fig. 2

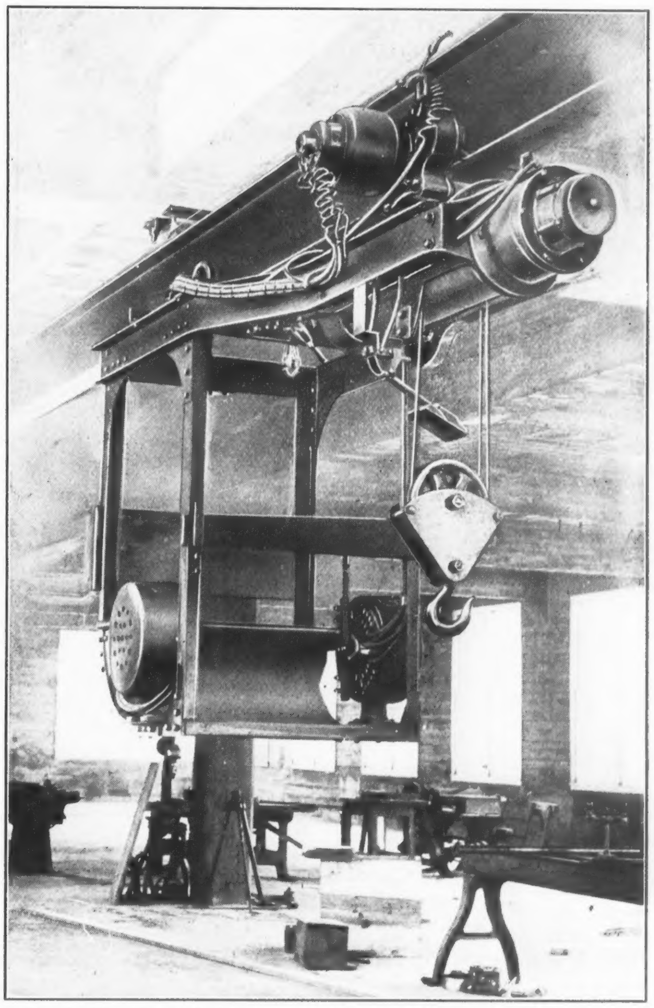
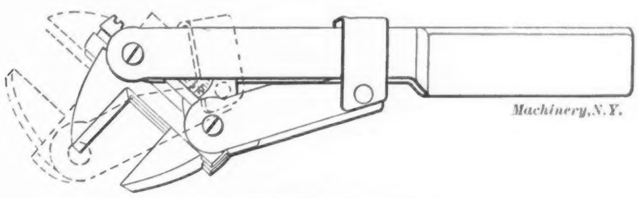


Fig. 2. The Shepard Hoist arranged with Self-traversing Trolley and Operator's Cab.

the device is shown applied to a trolley suspended from an I-beam as before, but with a second motor for power traverse, and a suspended operator's cab carrying the controllers, operated in this case directly by hand levers. The equipment shown in the latter figure is in use in the works of the American La France Fire Engine Co. Of course this hoist may be used in many other ways besides in the two applications shown.

RATCHET MONKEY-WRENCH.

The engraving shows a wrench of new and unusual construction which combines the handiness and quick adjustment of the monkey-wrench with the ability of the ratchet wrench to work in confined quarters. The jaws are adjusted to the size of the nut or screw head by a knurled thumb nut as in



An Adjustable Ratchet Wrench.

the monkey-wrench. As may be seen, the head containing these jaws is pivoted at one end to the handle, and at the other to a link connecting with a stirrup sliding on the handle. It may be used as a monkey-wrench with the jaws in the position shown if there is room for the turning movement re-

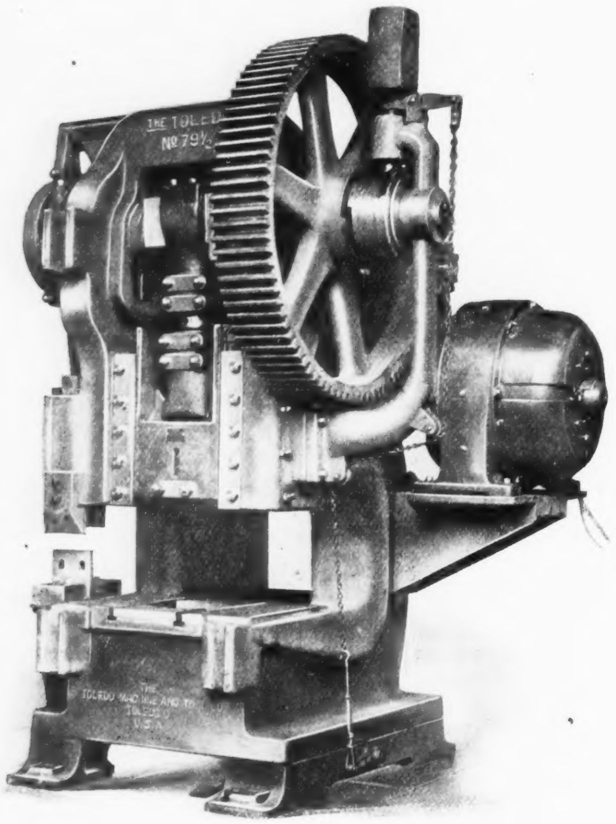
quired. When there is not room for this, the nut or bolt head may still be turned by successive short angular movements of the handle. The stirrup in this case acts as a ratchet, pushing the jaws around farther and farther with each movement. The head of the wrench can also be turned in the same way at any convenient angle, to get it into odd corners where an S-wrench would otherwise be necessary.

This tool is made by the Simplex Mfg. Co., 90 West Street, New York City.

LARGE OPEN-BACK SINGLE PITMAN GAP POWER PRESS.

The machine shown herewith was designed and built by the Toledo Machine & Tool Co. of Toledo, Ohio, for use in one of the United States Navy Yards for trimming hot or cold forgings. It is also intended for heavy blanking and shearing, thus making a tool of considerable range of usefulness.

As shown, the frame is of the over-hanging or gap pattern, with provisions for using the rods at the front for unusually heavy work which does not require the use of the gap. It is powerfully geared, and has an unusually long stroke for the main ram (7 inches), adapting it to the trimming of forgings of irregular or special shapes. An outer or shearing slide is provided on the left-hand housing. This has a stroke of 3 inches, and is capable of shearing, hot, 3-inch square bars.



Press for Heavy Trimming, Blanking and Shearing.

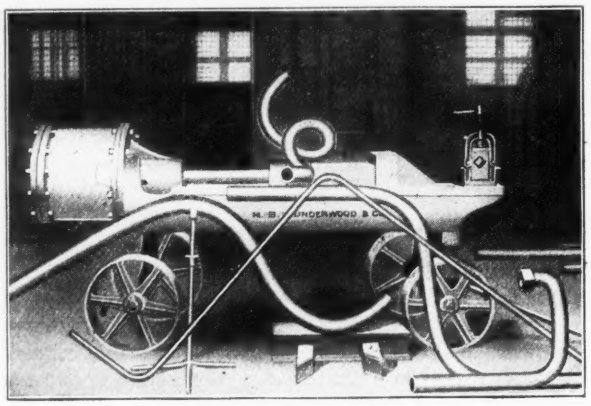
The clutch gear of this machine is 62 inches in diameter by 10 inches face, and the balance wheel, which weighs 1,600 pounds, is 50 inches in diameter. The distance between the uprights is 27 inches, the distance from the bed to the slide is 20 inches, and the area of the bed is 41 inches from right to left, and 28 inches from front to back. The distance from the center of the slide to the back is 16½ inches. The net shipping weight of the machine is 35,000 pounds. This press is the largest of a line of fifteen sizes designed and built by the makers.

RIGHT-ANGLE DRIVE FOR DILL SLOTTER.

The T. C. Dill Machine Co., Kensington, Philadelphia, Pa., has recently built a number of slotters with a drive at right angles to the axis of the base. This arrangement, which is believed to be new in slotter construction, has been found to be of marked advantage in allowing the tool to be placed in the proper relation with the line-shaft and the apparatus for placing and removing the work. This is especially true in the case of shops of the usual construction for heavy work, having a central bay served by a traveling crane, with side bays provided with line shafting, running lengthwise of the shop. Under these conditions, the slotter can be placed with the table projecting out from under the gallery into the space served by the crane, in which case its driving shaft will be parallel with the line-shaft from which it receives its power.

UNDERWOOD PNEUMATIC PIPE BENDING MACHINE.

H. B. Underwood & Co., 1024 Hamilton St., Philadelphia, Pa., have recently added to their list of tools a pneumatic pipe-bending machine, covering the range required in the ordinary work of the locomotive or air brake repair shop. It is

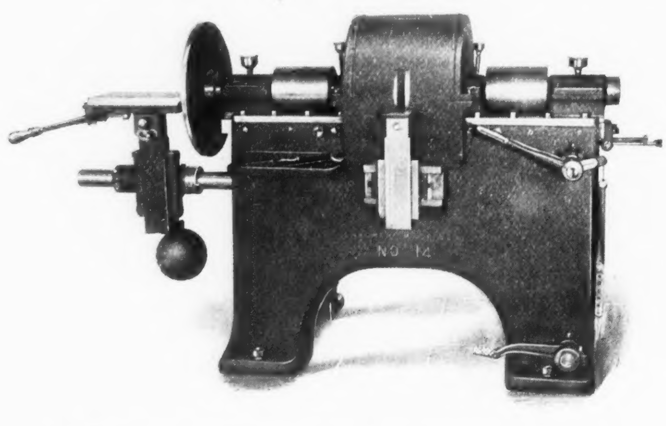


A Pneumatic Machine for Bending Pipe.

pneumatically operated, and will bend pipes to the desired radius without the filling and heating required for hand-operated machines. It will make a right-angle bend in a 2-inch pipe in two minutes, without flattening or injuring the work in any way. It will handle pipe from ½ inch up to 2 inches in diameter, in any standard radius required for locomotive work. Special dies for special radii or shapes are made to order.

COMBINATION SINGLE AND DOUBLE DISK GRINDER.

The machine shown herewith is a combination single and double disk grinder built by the Gardner Machine Co. of Beloit, Wis. It consists of a rigid box frame provided with ways on the top for two spindle heads, of which the one at the right is a sliding head operated by hand or foot lever, while that at the left is adjustable through a range of 8 inches.



Gardner Disk Grinder arranged for Single and Double Surfacing.

The latter head also carries a disk at each end of the spindle, the outer one being adapted for the full range of ordinary disk grinder work. For this work a table of any style desired may be furnished, the one shown being what the makers call their "universal lever feed table," adapted for a large variety of manufacturing operations. The feeds of this table are provided with micrometer stop screws on all movements.

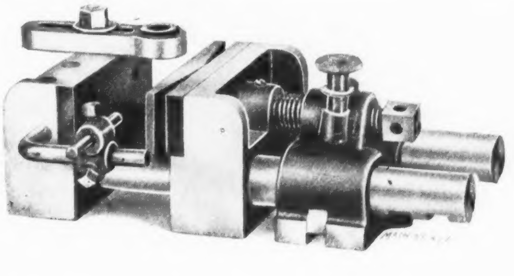
The double head grinder is adapted for machining rough pieces having equal opposite parallel faces, such as washers,

nuts, bolt heads, etc. The wheels are covered by a cast iron hood provided with a sheet steel shield, in which an opening of any desired size may be cut to suit the work being operated on. When an exhauster is used, this hood so encloses the machine that the operation is as dustless as that of turning or milling. As may be seen, the work is supported by a narrow table, adjustable on ways at the front of the machine for height and lateral position. This table projects between the two disks and supports the work when the disks are pressed against it. Micrometer stop screws are provided for limiting the movement of the heads toward each other, so that the work may be finished accurately to size. In work which requires both the hands of the operator for holding it, the feeding pressure is applied by the foot lever shown, a retracting spring being provided for withdrawing the movable head when pressure is removed, while a back stop limits its retracting movement.

The spindles of this machine are very large, and are driven by 8-inch belts. The end thrust is taken on large hardened steel collars bearing on composite babbitt and cast iron thrust surfaces. Double disks of any size up to 20 inches, and an outside disk up to 23 inches in diameter, may be used if desired. Chucks for carrying ring emery wheels are provided by the builder if the purchaser desired to use these on certain classes of work. A full assortment of tools and equipment, such as setting up press, cement, wrenches, etc., are provided with the machine.

THE BOWN DRILL PRESS VISE.

The drill press vise, shown in the accompanying cut, has been designed with the object of overcoming the inconvenience of hunting for straps, clamping bolts, nuts, etc., every time a piece of work has to be clamped to the drill press table. A great deal of time is spent unprofitably, because of lack of proper provisions for holding work on the tables of machines; and, although it is difficult to say to what extent the breakage of drills is due to improper means of holding work, it is probable that the expense incurred in this respect cannot be lightly considered. With a drill press vise, such as is shown in the cut, the work can be clamped instantly. As will be seen from the illustration, the vise is also provided with an adjustable strap, containing a drill bushing,



A Drill Press Vise with Locating and Jig Bushing Attachments.

and with adjustable stops, so that interchangeable work can be drilled in the vise accurately. The saving resulting from this use of the vise is apparent, as the expense of drill jigs for many ordinary jobs becomes unnecessary. In cases where a drill jig would not be made, then a great deal of time would have to be spent in laying out each piece of the work for drilling; this also becomes unnecessary for simple duplicate drilling when the vise is used. When work with many holes is to be drilled, a jig plate can be placed between the jaws, and this jig plate, covering the work, can contain any number of holes desired. In order to hold taper or irregular work rigidly, one jaw is made to swivel. Jaws are also made provided with vertical grooves, for holding round pieces in a perpendicular position for centering or drilling. The vise may, of course, also be used on other machines, such as planers, shapers, or milling machines.

The movable jaw is instantly adjusted to any desired opening, by pressing down a knob, which is connected with the tool steel locking bar. By giving the knob a slight turn, the bar is held down until the jaw is moved to the right position. Turning the knob back again, allows the locking bar to spring back into the square notches milled on the lower side of the rods; these notches are plainly shown in the cut. The jaws are made of tool steel and hardened, and the rods are also made of tool steel and ground.

The width of the jaws is $5\frac{1}{2}$ inches, and the full depth to the bottom, $3\frac{1}{2}$ inches. The depth to the rods is $1\frac{3}{4}$ inch. The regular vise can be opened up to 8 inches between the jaws, but longer rods are made to order so that openings of any width can be obtained. The weight of the vise is 33 pounds. It is made by the Bown Machine Co., Ltd., Battle Creek, Mich.

CORRECTION TO DESCRIPTION OF PLANER CLAMP.

An error was made in our description of the J. H. Williams & Co.'s planer clamp, which was illustrated in the December issue. The clamp is not case-hardened as was stated therein, but is given a special heat treatment which toughens and strengthens the forging, making it better suited to stand the rough usage that such machine tool accessories commonly get.

* * *

Some experiments regarding the strength of cast iron beams have recently been carried out at the Worcester Polytechnic Institute, and the data obtained during these investigations are given in the November issue of the *Journal of the Worcester Polytechnic Institute.* The experiments were carried out by Messrs. E. A. Adams, C. S. Frary, and E. H. Fish, of the Polytechnic Institute. The conclusions arrived at are very interesting, more particularly so as they do not fully agree with the commonly accepted ideas that a much higher stress should be permitted in cast iron beams on the compression side than on the tension side. It will be recollected that in our November issue we published an abstract from a paper on physical characteristics of cast iron, in which this question was referred to in connection with the design of shear or punch frames of the open jaw or gap type, in which it was stated that the breakages on the tension side were often caused by comparatively too little metal being deposited on the compression side. While the report of the tests at the Worcester Polytechnic Institute concludes with the statement that it is difficult to arrive at a definite conclusion of the tests made, it is added that it appears to be correct to design beams on the assumption that the strength of the material is the same in tension as in compression. A design made in this way would seem to be wasteful of material on the compression side, but it is said if the indications from these experiments are correct, then a smaller factor of safety might be used than has previously been the custom. The investigators sum up this conclusion, however, with the statement that what has been found from the present experiments can be taken only as an indication, and that tests on a still wider range of cross sections and sizes would be necessary before they would consider that this statement could be accepted as an actual fact. It will be understood, of course, that, while in designing a beam to carry a load the design would be based on the same strength of material for tension as for compression, that should not be construed to mean that cast iron when subjected simply to stress or simply to compression has got the same strength in both cases. Of course the well-known relation between cast iron's tensile and compressive strength, each considered by itself, which was thoroughly investigated over a century ago, is as true to-day as ever.

* * *

An item in the *Daily Telegraph*, London, mentions that lately about 0.1 ounce of radium was produced at the University laboratory at Vienna. Small as this quantity is, it is claimed to be the greatest yet produced, and to obtain it, ten tons of uranium pitchblende had to be used. The process of obtaining this amount of radium cost $10,000. Important experiments will now be begun at Vienna to ascertain whether the theory of Sir William Ramsay, that radium can be converted into other elements, especially into helium and lithium, be correct or not. It is also mentioned that the Vienna Academy of Science, in recognition of the great services rendered by the English scientist in connection with radium discoveries, will lend him a fraction of this one-tenth of an ounce for the purpose of experiments.

INDUSTRIAL NOTES FROM EUROPE.

BRITISH TRADE TOPICS.

Though the aggregate of British trade keeps up wonderfully, signs are not wanting that considerable caution will, in many instances, be needed as regards future productive policies. German producers of iron and steel in various forms are not so well engaged as for some time past, and doubt as to the future action of the Continental and American syndicates tends to restrict British trade movements. The high bank rate, largely due to the drain of gold on account of American financial troubles, also considerably hinders current enterprise.

Trade Disputes and Labor Questions.

The threatened railway trouble has been obviated, largely through the tact and administrative ability displayed by Mr. Lloyd George, president of the Board of Trade. The agreement reached holds good for at least six years. Committees of various grades of railway men will confer with the local railway companies' officials with respect to any grievances. If the matter is not settled to the satisfaction of the men, the directors of the companies may be negotiated with. Failing a settlement, the matter is to be placed before a Conciliation Board composed of equal numbers of representatives of the companies and the men, the men's trades union officials being eligible as members of the board; and an arbitrator appointed by the Speaker of the House of Commons or the Master of the Rolls, or both, will give the final decision, which will be binding on both parties. Though deprecated by extreme trade unionists, the settlement arrived at is generally approved as admitting the principle of trades union recognition, with a minimum of interference with the internal discipline of the railways.

Ship Building and Naval Topics.

Shipbuilding returns show a continued decline, both on the northeast coast and on the Clyde, new orders comparing very unfavorably with the corresponding period of last year. The rapid completion and fine trial records of the *Mauretania* are naturally very gratifying to the Northeastern district, its constructional facilities being thus proved second to none. In naval circles current matters of interest include the launch of the *Cyclops*, now being fitted at Portsmouth. The vessel will be the most powerful and completely equipped repair ship afloat. The lifting shears fitted on board will be capable of lifting a torpedo boat completely out of the water. The naval dockyard extensions at Devonport are of an important character. The new crane will lift 160 tons at a radius of 95 feet, 80 tons at 115 feet, and 30 tons at 128 feet, and is understood to be the largest of its class yet installed. Satisfactory progress is being made with the piecing up of the *Suevic*, and it would appear, as a consequence of precautions taken, that the vessel will be more seaworthy than when new.

General Progress of British Industries.

During the last few years British manufacturers have, in the face of international competition, done so well generally, in consequence of adoption of meritorious foreign methods or machinery, that there is some little danger of a return of the self-satisfied condition which preceded the temporary setback experienced in some branches about ten years ago. Mr. Joseph Adamson, president of the Manchester Association of Engineers, in his presidential address, drew attention to the importance of the Manchester district retaining, or regaining, its previous reputation as a center of industrial research and pioneer work. It is only necessary to mention such names as Crompton, Arkwright, Roberts, Nasmyth, Whitworth, Bodmer, Fairbarn, Daniel Adamson, Joule, Dalton, etc., to indicate the type of men who initiated the characteristic productions and influence of Manchester engineering, and the task which awaits their industrial successors. Mr. Adamson was of the opinion that standardization of designs, products, and methods, had gone far enough for a while, and that more attention should be paid to original work than has been the case for some time. Curiously enough, similar advice has recently been proffered to American manufacturers and engineers.

The utilization of what has often been considered waste heat or gases from blast furnaces, etc., is being much more systematically carried out than formerly. The latest proposition, likely to be carried into effect, is to employ the waste heat from coke ovens for the production of electric current, the plant being run by special power companies, who will purchase the gases from the coke oven proprietors and sell the current. In some instances the same procedure will be followed in the case of blast furnace gases.

Touching on South America and neighboring machinery business, it is of interest to note the obtaining by Mirrlees, Watson & Co., of Glasgow, of a large contract for the machinery of a sugar factory at Porto Rico, the order, which amounts to about $350,000, being obtained against considerable international competition.

Automobile Industry.

The outstanding feature at the motor car exhibition at Olympia is the strong position of British makers, who appear to have quite made up the considerable leeway which divided them from their Continental competitors a few years back. The greatest successes of the Continentals have been represented by the cars constructed practically regardless of expense, but in consequence of the bad summer and overproduction, the firms building these cars are not so well placed as the British builders mostly are, who, while offering the expensive styles, have also planned to easily change, if necessary, to the manufacture of moderate, or even comparatively low-priced, cars, and in fact the latter are now being made a leading line by several makers. While special tools for manufacturing the parts of motor-car mechanism have been designed and largely sold, the needs of repairers and small makers are receiving a fair share of attention. As was the case a few years ago in the cycle trade, small makers can often produce a small or medium priced car of a quality to compare favorably with the product of the large concerns, and at a distinctly lower price.

Machine Tools.

A lathe of 9-inch swing, adapted for repair and amateur workshops, is being built by King & Co., Armley, Leeds. In addition to turning and boring work, it is easily adaptable for many milling jobs. The main variations from ordinary lathe design are in the saddle and tail-stock. The saddle slides on a pair of Vs, cast on the front face of a strong bed, and carries a vertical pocket, the bore of which corresponds to the bore of the tail-stock barrel. Cast onto the under side of the compound slide-rest is a shank which is turned to fit both the saddle pocket and the tail-stock barrel, a nut in the end of the shank being made to engage with the screw in both barrels and saddle pocket. The change from a lathe to a milling machine is easily effected by taking out the spindle from the tail-stock, and then taking the compound rest out of the saddle pocket and putting it into the tail-stock barrel. The top slide of the compound rest consists of a table 8 x 5 inches, provided with two T-slots on which to secure tools and jigs when used as a lathe, and jigs and work when used as a milling machine. The table has a traverse of 10 inches, and a cross traverse of 5 inches. Within the limits of these traverses objects can be milled with practically the same facility as on an ordinary milling machine. Spur wheels up to 10 inches diameter by 1½ inch face can be cut, or greater widths up to 7 inches diameter. Attachments are in preparation for converting the lathe into a shaping and boring machine. Though, as a rule, machine tools of too comprehensive a range are not desirable, a tool as described can be profitably used in many cases. We may add that the machine, as described, is listed at $150 when arranged on a 4-foot bed. At one time the small upright and sensitive drill presses were almost an impregnable line with certain American makers. Though a good many of the best makes from the States are still sold, several makers over here find that they can make them with profit. In fact, one firm—Jones, Pollard & Shipman—almost confine themselves to upright and sensitive drills, and, finding their works accommodation altogether too cramped, are now building a large new works at Leicester, adjoining their old premises. With the additional equipment to be installed, the firm will be placed in an exceptionally good position for profitable manufacture and rapid output. JAMES VOSE.

Manchester, England, Dec. 5, 1907.

MISCELLANEOUS FOREIGN NOTES.

TRADE CONDITIONS IN ITALY.—Consul James E. Dunning, of Milan, advises that the reports of labor difficulties in Italy have been greatly exaggerated in the United States, and that American shippers should not withhold consignments from Italy on account of these rumors. There is at present a temporary depression in the money market, but general solvency is not affected. Traffic congestion has been common at all entering points and large centers, but this has been caused not so much by any unfavorable conditions of the railways, but rather by the present enormously increased industrial development of the country.

AUSTRALIAN TRADE CONDITIONS.—In a report from a Melbourne firm to the president of the Melbourne Chamber of Commerce, the present trade conditions of Australia were very thoroughly discussed, and from this report we have taken the following statements of interest to machinery manufacturers. Whereas ten or twelve years ago the whole of this firm's machinery came from England, to-day 60 or 70 per cent is supplied by America and Germany. American machine tools have in some branches entirely replaced the English, and at the present time some German machines have gained favor because of their superiority to the English and American designs. The trade of small machinists' tools has until very lately been in the hands of American manufacturers, but of late British makers have been competing successfully with high speed steel drills. In files, America undoubtedly holds the market. It is not contended, however, that the American files are better than English makes, but they are cheaper for the same quality. In small general tools, the Germans almost entirely supply the market, and it is stated that the success of the Germans in obtaining a trade in these goods in competition with American goods is due almost entirely to the superior German packing. If as little as proper packing can increase American trade in Australia, that is a matter which it seems could be easily remedied.

* * *

FIRST HARTFORD INDUSTRIAL SHOW.

The First Hartford Annual Industrial Show was held December 16 to 21 in Foot Guard Hall, Hartford, Conn., and was an undertaking which undoubtedly reflects much credit on its promoter, Mr. R. B. Jacobs, of the Jacobs Mfg. Co., Hartford, and his assistants. The attendance was large, the hall often being crowded with manufacturers, superintendents, shop managers, mechanics and others legitimately interested. While the first show was incomplete in general representation, on account of the short time given to advertising it and soliciting exhibits, it was a success. Hartford is a city well located for a mechanical and industrial exhibition, and there are indications that the show for 1908, the plans for which are already being made, will be on a much broader scale. The booth arrangements were well worked out. The signs and decorations were in good taste, some of them being of a very attractive nature.

Following is a list of exhibitors and exhibits, in the machinery line only:

Edwin B. Bartlett, Boston, Mass. Arbor presses.

Billings & Spencer Co., Hartford, Conn. Drop forgings and tools.

Bullard Automatic Wrench Co., Providence, R. I. Line of Bullard automatic wrenches.

Carlyle Johnson Machine Co., Hartford, Conn. Johnson standard friction clutches and reverse drive clutch for direct driving from line-shafts.

Cedaroleum Co., Perkinsville, Vt. "Cedaroleum" anti-rust compounds.

Cling Surface Co., Buffalo, N. Y. Model demonstrating the effectiveness of "Cling Surface" belt dressing.

Colton Combination Tool Co., Easthampton, Mass. Combination lathe tool-holders.

H. C. Cook Co., Ansonia, Conn. Sensitive drill and double-head drilling lathe.

Garlick Saw Co., Meriden, Conn. Flexible back "Boss Cutter" hack-saw blades.

Henry & Wright Mfg. Co., Hartford, Conn. Multiple-spindle sensitive drill presses.

Jacobs Mfg. Co., Hartford, Conn. Drill chucks.

Lancaster Machine & Knife Works, Lancaster, N. Y. 18-inch engine lathe fitted with the Derrer universal shape attachment for turning and boring eccentric, oval, square and multi-lobed shapes and for relieved taps and millings cutters.

Lincoln-Williams Twist Drill Co., Taunton, Mass. Twist drills.

Massachusetts Saw Works, Chicopee, Mass. "Victor" hack-saws.

O. K. Tool Holder Co., Shelton, Conn. Line of lathe, shaper, planer tools and drop forgings.

Pratt & Whitney Co., Hartford, Conn. Engine and turret lathes, cutters, reamers, twist drills, etc.

A. E. Quint, Hartford, Conn. Turret drill and tap holder.

Saxon Machine Co., Holyoke, Mass. Surface grinder.

Dwight Slate Machine Co., Hartford, Conn. Sensitive drill press, etc.

W. M. & C. F. Tucker, Hartford, Conn. Oil cups.

Union Twist Drill Co., Athol, Mass. Twist drills, reamers, milling cutters, etc.

Vitrified Wheel Co., Westfield, Mass. Line of vitrified emery and corundum wheels.

Waltham Machine Works, Waltham, Mass. Automatic pinion and gear-cutting machine.

Wells Bros. Co., Greenfield, Mass. Taps, dies and gages.

* * *

ARTHUR WILLIAMS HONORED.

A luncheon was given to sixty at the Engineers' Club, New York, December 18, in honor of Mr. Arthur Williams, general inspector of the New York Edison Co., at which he was presented with the decoration Officier de L'Instruction Publique. This decoration was recently conferred on Mr. Williams by France in recognition of his public service in general welfare work for the several thousand employes of the New York Edison Co. The luncheon was given under the auspices of the American Museum of Safety Devices and Industrial Hygiene, and was presided over by Mr. T. C. Martin, vice-chairman of the advisory committee. Dr. W. H. Tolman, the director of the Museum, made the presentation of the decoration, and, in a few words, told the reason for thus honoring Mr. Williams. Dr. Josiah Strong, president of the Museum, spoke of the general aspects of the present organized movement to prevent industrial accidents and to ameliorate the condition of the working classes in general. Mr. John Lieb, associate general manager of the New York Edison Co., in a short address outlined the policy of his company in this relation to its employes as regards their physical welfare. Mr. Williams' life work has been in the service of the New York Edison Co. He was born in 1868, and entered its service in 1885 as assistant in the meter department. His rise has been rapid.

* * *

PERSONAL.

A. M. Powell has sold his stock in the Woodward & Powell Planer Co. to E. M. Woodward, and has retired from the company.

Ulrich A. Peters, Pittsburg, Pa., mechanical engineer and contributor to MACHINERY, sailed to France December 28 for a stay of six weeks.

A. L. Lovejoy, treasurer of the Becker-Brainard Milling Machine Co., Hyde Park, Mass., resigned his position on December 31, and states that his plans for the future are not yet matured.

Ivy L. Lee has retired from the firm of Parker & Lee, New York, to enter the service of the Pennsylvania R. R. Co. He will have charge of the publicity work of the company, hitherto carried on by Parker & Lee.

Charles A. Bridge, hitherto manager of Parker & Lee, New York, has been made partner following the retirement of Ivy L. Lee. Dating from January 1, 1908, the firm will be Parker & Bridge.

J. P. Jackson, professor of electrical engineering, Pennsylvania State College, was recently appointed by the board of trustees of that institution to the position of dean of the school of engineering.

S. Owen Livingston, who for several years past has been with the Fox Machine Co., in charge of the advertising and the foreign sales departments, has severed his connection with that concern to become interested in the Wilmarth & Morman Co., Grand Rapids, Mich., manufacturers of drill grinders, etc. Mr. Livingston will take active charge of the sales and advertising for the Wilmarth & Morman Company.

The only reason why we need to advertise the

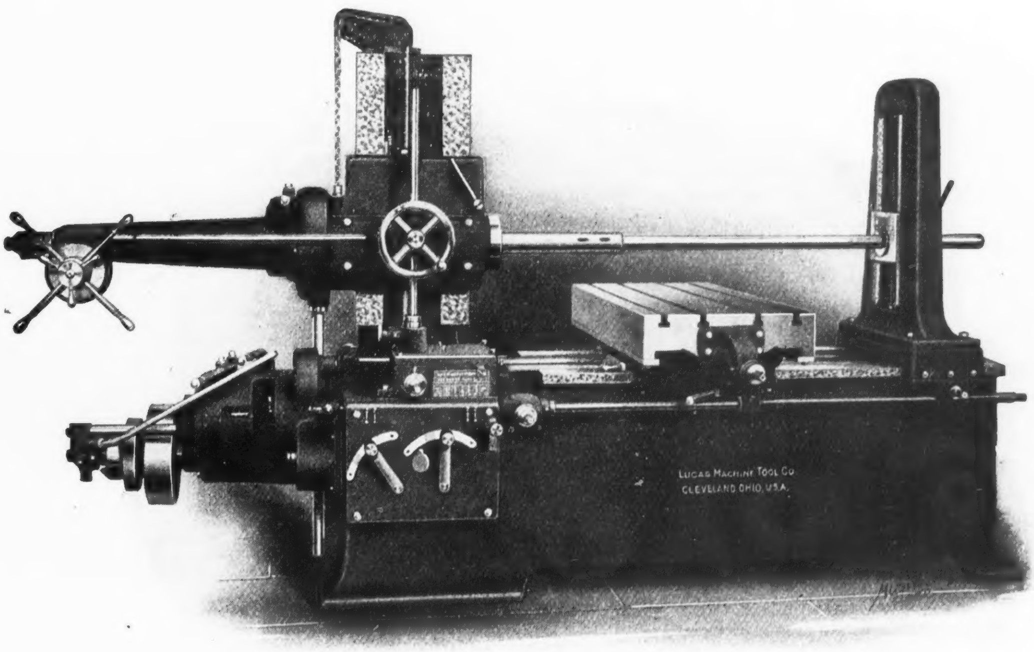
LUCAS (OF CLEVELAND)

"PRECISION"

Boring, Drilling and Milling Machine

is that there are *some* concerns that do not *yet* know about it.

We recently ran across one of these concerns; they had bought for contract work an ordinary boring machine *and a planer.*



This machine would have done the work quicker and better, would have done it at ONE SETTING, would have NOT taken up as much room, and would have cost less than the other two machines.

Lucas Machine Tool Company

CLEVELAND, OHIO, U. S. A.

European Agents: C. W. Burton, Griffiths & Co., London. Alfred H. Schutte, Cologne, Brussels, Liege, Paris, Milan, Bilbao, Barcelona. Schuchardt & Schutte, Berlin, Vienna, Stockholm, St. Petersburg.

OBITUARY.

William Cox, inventor of a great variety of slide rules, computers and other mechanical devices for facilitating engineering calculations, died December 1. The planing and shaping computer illustrated in our August, 1907, issue is an example of his activities in this line. This computer was the 120th produced.

Daniel T. Campbell, president of the Watts-Campbell Co., Newark, N. J., builder of Corliss steam engines, died December 16 of pneumonia. Mr. Campbell was connected with the concern for nearly fifty years, having entered into its employ as a clerk when a young man. He was admitted to partnership in 1865, and became president in 1884.

LORD KELVIN (SIR WILLIAM THOMSON).

Lord Kelvin, the great British scientist, died in Glasgow, December 17, at the age of 83. Lord Kelvin, or William Thomson, as he was known before being elevated to the peerage, was born in Belfast in 1824. Eight years after his birth, his father, James Thomson, was appointed professor of mathematics in the University of Glasgow, and the son became a student in the university at the age of 11. After the completion of his course in the University of Glasgow, young Thomson removed to Cambridge and graduated from St. Peter's College in 1845. The next year he was made professor of natural philosophy in the University of Glasgow, which position he held for 53 years, retiring in 1899. From his early youth he manifested a true scientific spirit. The first abstruse problems which attracted his attention was the figure of the earth and Fourier's theory of the flow of heat, and his contributions to these subjects attracted wide attention. The work which made him famous, however, was that of electrical engineer for the company that laid the first Atlantic cable. He was electrical engineer of the expedition that laid the first cable in 1857-8. At that time electrical engineering was in a primitive state, and the art of cable working was undeveloped beyond a very incomplete theory, Thomson invented a siphon recorder, which was used for many years to receive cable messages, and it remained up to recent years the most sensitive instrument of the kind ever devised. He made many electrical inventions which broadly may be divided into three great groups: First, those connected with telegraphy and submarine cables; second, those connected with navigation and the mariners' compass; and third, those connected with instruments of precision for electrical instruments. Lord Kelvin was an indefatigable worker, and continued profound research up to the time of his death. When he resigned his professorship at the Glasgow University, he still desired to be connected with the institution, and at the age of 75 he entered his name as a student! A volume might be written of his remarkable achievements, and it would by no means exhaust the record of work done by this remarkable man.

CHARLES P. MATTHEWS.

Charles P. Matthews, professor of electrical engineering at Purdue University, died at Phœnix, Arizona, November 23, 1907. Prof. Matthews was of Vermont stock, his family going from that state to New York in 1852, where, at Fort Covington, he was born September 18, 1867. At the time of his death he was, therefore, a little more than forty years of age. He attended the St. Johnsbury Academy at St. Johnsbury, Vermont, graduating in 1887. He then entered Cornell University, graduating from Sibley College, with the degree of Mechanical Engineer, in 1892. In 1901 he received the degree of Doctor of Philosophy from his alma mater. Immediately after graduation, he became instructor in physics and applied electricity at Cornell, serving in that capacity four years, until 1896. At that time he was called to Purdue, and was appointed associate professor of electrical engineering. In 1905 he succeeded Prof. Goldsborough as head of the school of electrical engineering, and from this time until his death he was continuously a member of the Purdue faculty.

During Prof. Matthews' connection with the Purdue school of electrical engineering, it has grown to be the largest in the country in point of numbers. In this development, he has had a large share. His instruction was of the highest order, not only on account of his professional ability and training, but quite as much on account of his exceptional personality and gifts. He made valuable contributions to his science, his chief work being an investigation of photometric standards for arc lamps. This was done in connection with the National Electric Light Association. In this, he directed all of the experimental work, designed the apparatus and prepared four reports aggregating about two hundred pages. In this connection he devised and patented an intergrating photometer. This instrument received a gold medal at the Louisiana Purchase Exposition. He was also collaborator in the production of text-books in physics and electricity with Profs. Nichols and Shearer of Cornell, and with Prof. Esterline of Purdue, and he had published a number of papers on electrical subjects. He was a member of the honor fraternity, Sigma Xi.

* * *

NEW BOOKS AND PAMPHLETS.

THE GAS ENGINE IN PRINCIPLE AND PRACTICE. By A. H. Goldingham. 195 pages, 6 x 9 inches. 105 illustrations. Published by the Gas Power Publishing Co., St. Joseph, Mich. Price, $1.50.

The work describes the various types of internal combustion engines, and compares the two- and four-cycle types. A chapter on valves and valve motion then follows, after which come governors, igniters, self-starters and other details; testing gas engines; use of the indicator and brake; representative indicator cards and defects shown by the indicator. Chapter VI. treats of the use of crude oils, fuel oils, distillate and illuminating oils. Notes on gas producers and gases are also included. The work is a fairly comprehensive treatise of the gas engine, although the matter given to any one subject is brief.

THE MECHANICAL WORLD POCKET DIARY AND YEAR BOOK FOR 1908. 391 pages, 4 x 6 inches. Illustrated. Published by Emmott & Co., Ltd., Manchester, England. Price, 6d. net.

This convenient and valuable year book contains matter on steam and steam engines, giving tables of speeds; mean pressure constants; properties of saturated steam; table of actual ratios of expansion; tables of indicated horse-power per pound of mean effective pressure; steam pipe diameters and port areas for various piston speeds and steam velocities; valve diagrams; short treatise on the steam engine indicator; proportions of marine crank-shafts; notes on condensing plants; notes on the steam turbine, with short description of various types; data on steam boilers and boiler setting, etc. Data are also included on gearing, belting, rope driving, both wire and fiber, friction clutches, bolts, nuts and screw threads, and furnaces. Useful mathematical tables and blank pages for diary and memoranda conclude the book.

TESTS OF REINFORCED CONCRETE BEAMS. Series of 1906. By Arthur N. Talbot. 36 pages, 6 x 9 inches. Published by the University of Illinois, Urbana, Ill.

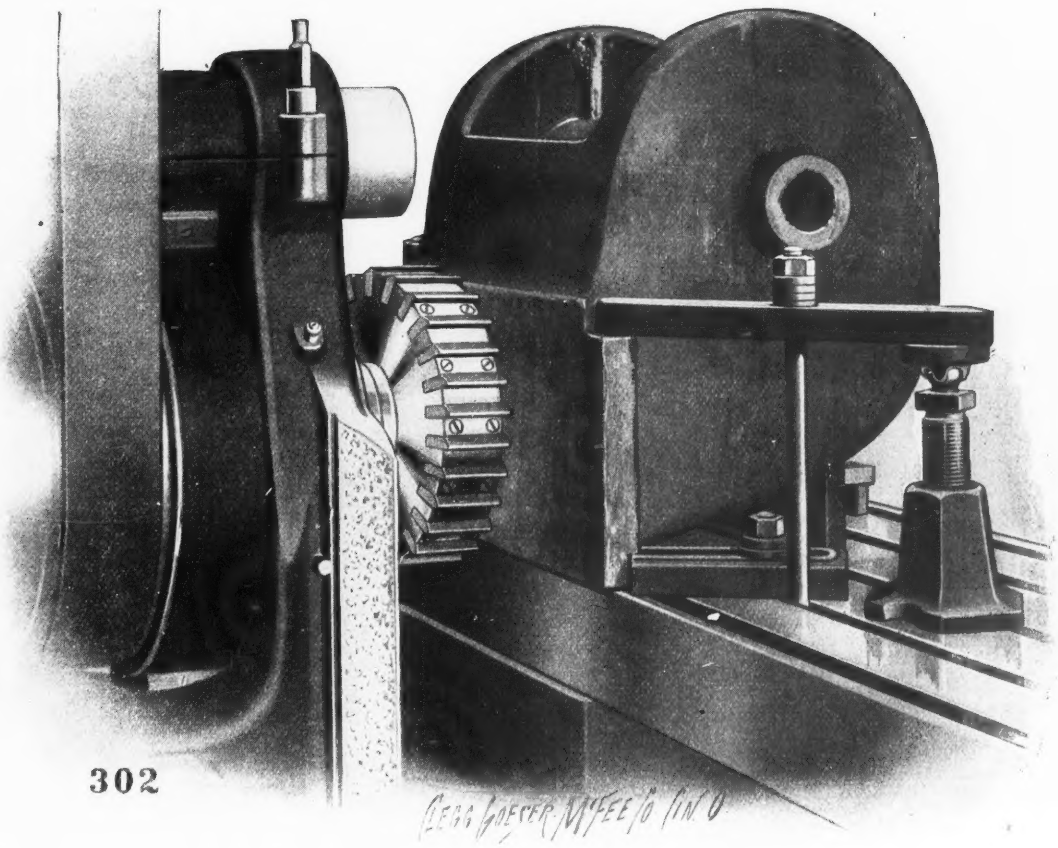
This is Bulletin No. 14, recording results of tests made by the University of Illinois Engineering Experiment Station. The tests described are a continuation of the tests discussed in Bulletin No. 4. The topics investigated include the effect of quality of concrete upon the strength of beams, the effect of repetitive loading upon the action of beams, and the resistance of beams to diagonal tension failures. The results of the investigation of diagonal tension failures throw light upon the amount of the vertical shearing stress which may be allowed in reinforced concrete beams not having metallic web reinforcement. The resistance of beams to diagonal tension may be the controlling feature of relatively short beams, and as such failures occur suddenly and without much warning, a knowledge of the resistance of the concrete is essential. Some beams gave surprisingly low values, and it seems evident that the values allowed by many city building ordinances are higher than should be recommended. The tests of concrete columns and reinforced concrete columns and of reinforced concrete T-beams for 1906 have already been published.

CARBURETING AND COMBUSTION IN ALCOHOL ENGINES. By Ernest Sorel. Translated from the French by Sherman M. Woodward and John Preston. 269 pages, 5 x 8 inches. 31 illustrations, plates and diagrams. Published by John Wiley & Sons, New York. Price, $3.00.

Now that internal revenue duties on denatured alcohol have been abolished by the United States, the possibility of using alcohol for internal combustion motors has created much commercial interest in it. The book before us is a treatise on the carbureter and combustion in alcohol engines. It treats of the records of tests of alcohol as an engine fuel; general and historical facts; conditions affecting gaseous mixtures; phenomena of combustion and gaseous mixtures; actual combustion in engines; carbureting; temperature of vaporization; investigations of carbureters, etc. In view of the scarcity of works on alcohol engines and the action of alcohol vapors, the work is one that should meet with general favor among engineers. The treatment is to some extent mathematical, but in the main it is well within the comprehension of the average person who would be interested in this phase of the internal combustion engine. It contains a great deal of valuable data heretofore unavailable save in foreign publications.

THERMODYNAMICS OF THE STEAM ENGINE. By Cecil H. Peabody. 533 pages, 6 x 9 inches. 117 illustrations, diagrams and charts. Published by John Wiley & Sons, New York. Price, $5.00.

This well-known work, which was first published in 1889 from notes prepared by the author for his classes in the Massachusetts Institute of Technology, has passed into the fifth edition, which is an indication of its permanent value. The fifth edition has been revised to bring it into accord with the development of recent engineering practice, particularly superheating and steam turbines. The author is still of the opinion that the general mathematical presentation due to Clausius and Kelvin is most satisfactory, but it is recognized that recent investigations of superheated steam are presented in such a way as to narrow the application of the general method so that there is good reason for the use of special methods. The revised work has been arranged with this fact in view, the general mathematical discussion being presented in a chapter by itself to be read or not as best suits the user. The introduction of the steam turbine has made adiabatic calculations for steam of practical importance, and columns of entropies of vaporization have been added to tables of properties of saturated steam. The contents by chapters are: Thermal Capacities; First Law of Thermodynamics; Second Law of Thermodynamics; General Thermodynamic Method; Perfect Gases; Saturated Vapor;



Fast Feed with Large Cutters

is one of the requirements of modern milling practice. Our No. 4 PLAIN MILLER with its immense spindle power and powerful feed drive meets these requirements. We submit the following as evidence:

> High speed cutter 12 in. diameter; taking a cut ⅛ in. deep across the entire surface of a grey-iron casting 9 in. wide, at nearly 5 in. per minute table travel. The piece is 18 in. long, and the cut takes just about 6½ minutes. The chucking takes about as long as on any other machine.

Figure how long this cut would take you on a planer. The difference between that time and 6½ minutes is what you will gain by milling if you will use a No. 4 Plain Cincinnati.

Let us tell you about this double back geared machine with all of its other recent improvements.

Ask us for time estimates with suggestions on milling your work.

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EUROPEAN AGENTS—Schuchardt & Schutte, Berlin, Vienna, St. Petersburg, Stockholm. Alfred H. Schutte, Cologne, Brussels, Liege, Milan, Paris, Turin, Barcelona and Bilbao. Chas. Churchill & Co., Birmingham, Manchester, Newcastle-on-Tyne and Glasgow. CANADA AGENT—H. W. Petrie, Toronto and Montreal.

Superheated Vapors; The Steam Engine; Compound Engines; Testing Steam Engines; Influence of the Cylinder Walls; Economy of Steam Engines; Friction of Engines; Internal Combustion Engines; Compressed Air; Refrigerating Machines; Flow of Fluids; Injectors; Steam Turbines.

CATALOGUES AND CIRCULARS.

T. B. Wood's Sons Co., Chambersburg, Pa. Catalogue No. 23, illustrating friction clutches, pulleys, etc.

Ferracute Machine Co., Bridgeton, N. J. Circular illustrating some of the many sizes and styles of Ferracute presses.

United Engineering & Foundry Co., Pittsburg, Pa. Catalogue of roll turning lathes made in 16-, 18-, 26-, 34-, 44- and 60-inch sizes.

United Engineering & Foundry Co., Pittsburg, Pa. Catalogue of high-speed saws for iron and steel works, including both hot and cold saws.

Buckeye Engine Co., Salem, O. Folder entitled "An Illustrated Talk on Halos," being an advertisement of the Buckeye electric blue-printing machine.

Goldschmidt Thermit Co., 90 West St., New York. Pamphlet on the Thermit Welding Process, illustrated with many examples of work done.

R. Woodman Mfg. & Supply Co., 63 Oliver St., Boston, Mass. Catalogue of railroad ticket punches, baggage checks and hat badges. A list of 444 dies, with illustrations of shapes cut out, is included.

Fosdick Machine Tool Co., Cincinnati, Ohio. Specification circular of the Fosdick No. 2 (Style D) horizontal boring, drilling and milling machine, which was illustrated and described in our September, 1907, issue.

Frank Mossberg Co., Attleboro, Mass. Catalogue No. 16, listing drop forged screw wrenches of the bicycle type, bicycle bells, advertising novelties such as paper cutters, etc., punches and dies, special machinery and screw machine products.

B. F. Sturtevant Co., Hyde Park, Mass. Bulletin No. 151, describing the Sturtevant steam turbine. The bulletin gives an interesting historical sketch of the steam turbine as a preface to the general matter. The Sturtevant machine is illustrated in detail, the details including the nozzles, bearings, governor, bucket wheels, vanes, etc.

General Electric Co., Schenectady, N. Y. Bulletin No. 4546, "The Electrification of the West Shore Railroad." The electrification of the West Shore R. R. between Utica and Syracuse is described, including the 60,000-volt transmission line, the inverted third-rail and the hydro-electric power development.

Fox Machine Co., 815-825 No. Front St., Grand Rapids, Mich. Catalogue No. 29 of Fox light milling machines, made in three sizes. This machine was developed to meet the requirements of typewriter manufacture, and is well calculated to meet the needs of manufacture of all light machine parts.

Mitts & Merrill, 843 Water St., Saginaw, Mich. New catalogue entitled The Giant Key-Seater, describing this machine and its parts in detail, and illustrating various other types of key-seating machines manufactured by this company. A section of the book is also devoted to key-seat milling machines.

Fox Machine Co., 815-825 No. Front St., Grand Rapids, Mich. Catalogue No. 67 of Fox tube and pipe cutters. These machines are of the rotary cutter type largely used in railway shops for cutting off boiler flues. They do not include threading attachments, but are confined to cutting-off work alone. The machines are listed in four sizes.

Fox Machine Co., 815-825 No. Front St., Grand Rapids, Mich. Catalogue No. 30 of Fox high-speed sensitive drilling machines made with one, two, three, four, five and six spindles. Each spindle has three changes of speed and independent belt tension adjustment. Other features of distinction are listed, including hardened and ground upper cone shafts, etc.

Kilbourne & Jacobs Mfg. Co., Columbus, Ohio. Circular of steel factory equipment, such as pressed steel pans, tote boxes, steel barrels, barrel trucks, etc. The pressed steel pans are made from one piece of heavy sheet steel. Having rounded corners and no rough surfaces whatever, they are well adapted for handling fine machine shop products without injury.

General Electric Co., Schenectady, N. Y. Bulletin No. 4537, "The Electric Locomotive in Heavy Passenger and Freight Work," in which is described a large number of present and proposed types of electric locomotives. Sketches are given of locomotives ranging in weight from 17 to 150 tons and adapted to mining, high-speed passenger, slow-speed freight, mountain grade trunk lines, etc. Valuable data are included.

General Electric Co., Schenectady, N. Y. Booklet giving standard symbols for wiring plans, adopted by the National Electric Contractors' Association and the American Institute of Architects, in a form that illustrates their use clearly. Sample plans of an office, kitchen and dining-room are shown with the electric wiring symbols to indicate the various appliances. The booklet also illustrates the General Electric specialties used in house and office electric installations.

Chandler Planer Co., Ayer, Mass. Bulletins M-3, M-4, M-5, M-6, M-7 and M-9 descriptive of the Chandler planers. Bulletin M-3 contains a bit of history of planer development and enlarges on the special features of the Chandler design, among which are case-hardened shafts, feed-regulating mechanism, change-speed mechanism, and finally and chief, high speed. Cutting speed is put at 30 feet for frog and switch planers, and ranges up to 90 feet for brass-working planers; the return stroke is made at speeds ranging from 100 to 150 or 200 feet per minute.

Cling Surface Co., 1018 Niagara St., Buffalo, N. Y. Booklet entitled "The Treatment of Belts and Ropes for Service and Profit," which gives the best methods of applying "Cling Surface" to leather, canvas, cotton, balata, link, camel-hair, rubber and rope belts. It also gives many valuable hints on the general treatment of belts. Illustrations of several typical belt drives on which "Cling Surface" is used, are included along with details of certain interesting tests. The booklet contains 87 pages and will be found of interest and value by anyone having the care of belting.

Hammacher, Schlemmer & Co., 4th Ave. and 13th St., New York. Circular descriptive of a portable machinist's tool chest which appears to be what many journeymen need. Its outside measurements are: Length, 15⅝ inches; height, 10⅜ inches; width, 8 inches. The drawers are felt-lined and run on hard wood slides. The front is provided with two hinged joints which permit it to fold back neatly and expose the drawers, of which there are three full length (14 inches) and five half length (6¾ inches). A flexible handle in the center of the top permits the chest to be carried the same as a suit-case.

Hess-Bright Mfg. Co., Philadelphia, Pa. Sheet, 9 x 12 inches, showing Hess-Bright ball bearing mounted for radial load without thrust. This sheet is the first of series entitled "Ball Bearings and Their Correct Use," which will show the principles of correct ball bearing mounting and scale drawings of various mechanisms in which ball bearings are used. The new series is an extension of the 3 x 5-inch index data card series issued by the company, which has met with much favor, but which, for the present purpose, is too small. The sheets are punched for a binder, thus permitting them to be gathered together in convenient form for reference.

Newton Machine Tool Works, Inc., Philadelphia, Pa. General catalogue No. 45 of machine tools, containing 303 pages and illustrating a great variety of special and standard machine tools. These include boring machines for bridge chords, cylinders, locomotive rods, motor frames and many other purposes; cold saw cutting-off machines; crank planers; drilling machines; gear cutters; milling machines; rotary planing machines; shaping machines; slotters; and miscellaneous tools not readily classifiable. The catalogue illustrates a number of the special floor-plate slotters, boring machines and drilling machines in use in the General Electric Works, Schenectady, N. Y., in which shops portable tool practice on floor-plates has reached a high development.

MANUFACTURERS' NOTES.

S. J. Smith Machinery Co. and L. Booth & Sons, Los Angeles, Cal., have consolidated as the Smith-Booth-Usher Co., with general offices and salesrooms at 212-214 South Los Angeles St.

Independent Pneumatic Tool Co. has moved its San Francisco office and storeroom from 11 Front St. to much larger quarters at 61 Fremont St., in the heart of the machinery district.

Rockford Drilling Machine Co., Rockford, Ill., is the successor to B. F. Barnes Co., of same city. The change is in the name of the company only, the place of business and product remaining the same as heretofore.

Wile Power Gas Co., Cleveland, Ohio, manufacturer of producer gas installations for power and fuel, has removed its general offices from the Cutler Bldg., Rochester, N. Y., to its works, 1688-92 Columbus Road, Cleveland, Ohio.

Gardner Machine Co., which has its main office and factory in Beloit, Wis., has opened a branch office in Cleveland, Ohio, 505 Caxton Building. The branch office will enable the company to take care of its increasing Central and Eastern business.

American Museum of Safety Devices and Industrial Hygiene, 231 W. 39th St., New York, announces that a friend of the Museum has offered a prize of $100 for the best essay on the "Economic Waste of Accidents." The committee of award consists of Richard Watson Gilder, George Gilmour and W. H. Tolman.

Coes Wrench Co., Worcester, Mass., announces that owing to the numerous requests for its 1908 souvenir, "A Vestal Virgin," by Sichel, it has had a limited number printed in Germany in full colors without calendar attached, size 20 x 29 inches, which will be sent, one only to an individual, on receipt of 40 cents to cover cost of wrapping and postage.

The name of the concern mentioned in the "Manufacturers' Notes" appearing in December as being the builder of the new Libbey turret lathe should have been the International Machine Tool Co. instead of the Indianapolis Machine Tool Co., as it erroneously appeared. The company is erecting a new factory and expects to be in full operation this month.

Peter A. Frasse & Co., New York, have opened a branch warehouse at 408 Commerce St., Philadelphia, Pa., for the sale of Shelby steel tube, Poldi tool steels, drill rod, music wire, Poldi high-speed drills, etc. This branch warehouse will handle only such specialties as the above and will not carry the full line of supplies to be had at their New York store, 94 Fulton St.

Hartford Blower Co., Hartford, Conn., recently installed a very large dust collecting system for Comstock, Cheney & Co., Ivoryton, Conn., for handling shavings, saw-dust, etc. The total height of the collector is 35 feet 5 inches, and the weight of the material used was 13,287 pounds. There are three discharge pipes, two being 40 inches diameter and the third 52 inches diameter. This installation is the fourth that the Hartford Blower Co. has made for this concern.

A. H. Briggs, manager of the New York store of the L. S. Starrett Co., Athol, Mass., since its opening ten years ago, has resigned to go into business for himself. He will be associated with Mr. W. M. Briggs, New York manager for the Rock Island Tool Co., Rock Island, Ill., manufacturers of vises, and the Joyce Cridland Co., Dayton, Ohio, makers of jacks. The firm will represent manufacturers of first-class lines of tools, etc.

S. Obermayer Co., Cincinnati, Ohio, is sending two slips with all letters, entitled "Put Your Shoulder to the Wheel—NOW," and "To the Man who Signs the Checks," entreating all to pay their bills promptly, and asking the recipients of checks from the S. Obermayer Co. to keep the ball rolling. In brief, the slips make a sensible and patriotic appeal to stop hoarding and to help business by paying all bills promptly.

G. M. Yost Mfg. Co., Mechanicsburg, Pa., is building a new factory at Meadville, Pa., to which place the company will move both the manufacturing end and business offices about March 1. About $5,000 worth of new machinery will be added to the equipment. The new plant will be organized and equipped for the manufacture of vises of all kinds and for all purposes, and will be one of the best plants in America. The business depression has not affected the company's business to an appreciable extent so far, and it regards the future in the most optimistic spirit.

Pittsburg Automatic Vise and Tool Co., Pittsburg, Pa., has recently patented an improvement, which it is applying to the jaws of its line of vises. This improvement consists simply in cutting a half-round groove of a radius of about ⅝ inch, lengthwise of the face of one of the jaws of the vise. The purpose of this groove is to permit the holding of round bar stock, tubes, etc., with a much greater degree of firmness and precision than is possible with flat gripping surfaces. In wood-working vises, the groove is of considerably larger diameter to agree with the class of work for which it is intended.

Independent Pneumatic Tool Co., Chicago, Ill., held a meeting of the stockholders in Jersey City, N. J., recently, at which the officers were re-elected, as follows: James B. Brady, president; W. O. Jacquette, first vice-president; John D. Hurley, second vice-president; A. B. Holmes, secretary and treasurer. The annual statement shows a large increase of business over the previous year. The company has greatly enlarged its Aurora, Ill., plant, and is in good financial condition. It is optimistic as regards future business. No cancellations of orders for "Thor" pneumatic tools have been received, nor has the company cancelled any of its orders for materials or machinery.

Curtis & Curtis Co., 8 Garden St., Bridgeport, Conn., which for the past twenty-five years has been engaged in the manufacture of pipe cutting and threading machinery, has just completed a large addition to its plant. The roof of one of the main buildings was raised, thus providing 8,000 square feet additional space for the machine shop and offices. The increased space was badly needed, the company for some time past having found it difficult to meet the demands for the Forbes die stocks, etc. The increased space will enable the manufacture of this and other specialties to be carried on in a more effective and economical manner. The increased office space has been fitted up correspondingly to properly care for increased business.

American Museum of Safety Devices and Industrial Hygiene, 231 W. 39th St., New York, announces that Prof. F. R. Hutton, past president of the American Society of Mechanical Engineers, is the chairman of the committee on admission of exhibits for the Museum. The Museum desires exhibits of devices and processes of safeguarding life and limb in connection with woodworking machinery, railway and marine transportation, mining, agriculture and manufacturing of all

CONTENTS FOR JANUARY.

kinds. The Museum occupies the entire fifth floor of a modern building, and has facilities for showing both dead and live exhibits to the best advantage. All exhibits accepted by the committee on exhibits will be eligible for the gold medal offered by the *Scientific American*. This medal is to be given for the best device exhibited for safeguarding life and limb. All inquiries regarding conditions should be directed to Dr. W. H. Tolman of the Museum.

COMING EVENTS.

January 10—Annual Meeting of the Central Railway Club, at Buffalo, N. Y. Harry D. Vought, 62 Liberty St., New York, Secretary.

January 14—Monthly evening meeting of The American Society of Mechanical Engineers in assembly room No. 1, of the Engineering Societies' Building at 29 West 39th St., New York. The subject will be "Car Lighting," the presentation being made by Mr. R. H. Dixon, president of the Safety Car Heating and Lighting Company, and will treat of the general subject of lighting railway cars, showing relative economies in the several systems, electric and gas. There will be in operation exhibits of different methods such as the Pintsch mantle, the vapor mantle system, a new acetylene system, and several varieties of axle lighting by electricity with their regulating and governing mechanism. Each member may bring one friend.

January 14 to 18—Automobile Dealers' Show, Foot Guard Armory, Hartford, Conn.

January 15—Annual Meeting, American Society of Civil Engineers. Charles Warren Hunt, 220 West 57th St., New York, Secretary.

January 23-25.—Annual Meeting, National Society for the Promotion of Industrial Education, Chicago, Ill. C. R. Richards, Columbia University, N. Y., secretary.

February 20 to March 7—Fourteenth Annual Motor Boat and Sportsman's Show, Madison Square Garden, New York City.

March 7 to 14—Sixth Annual Boston Automobile Show, Mechanics' Building, Boston.

MISCELLANEOUS.

Advertisements in this column, 25 cents a line, ten words to a line. The money should be sent with the order. Answers addressed to our care will be forwarded. Original letters of recommendation should not be enclosed to unknown correspondents.

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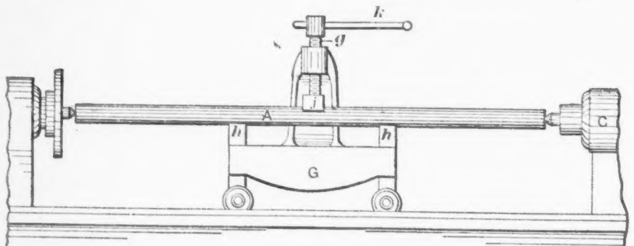
SHOP OPERATION SHEET NO. 52.

H. K. Griggs. MACHINERY, February, 1908.



To Straighten a Rough Bar for Making a Shaft.

1. Take the rough bar A to the centering machine, and drill and ream center holes in each end.

2. Place the shaft straightener G on the lathe bed with its supporting wheels between the inside and outside vees.

3. Fasten a lathe dog to one end of the bar A, and place the bar between the lathe centers.

4. Start the lathe at a moderate speed, the slowest speed with the gears out being about right, and hold a piece of chalk close to the bar. If the bar runs out of true, the chalk will make a mark on the high side. In this way test the bar at various points until the greatest bend has been located.

NOTE.—When straightening a small bar, it may be rotated by drawing the hand rapidly across it. This will cause the bar to revolve on the centers, and then the chalk may be held against it as described. When rotating a bar in this way, the dog is, of course, not needed.

5. Turn the bar A until the chalk marks are up, and then move the straightener G to the most pronounced bend in the bar, as indicated by the chalk marks. Bring the pressure-screw g directly over this bend, and place the blocks h under the bar as shown. Slack up on the lathe centers to avoid springing them. Bring down the pressure-screw block j upon the bar A, and with the lever k turn the screw g sufficiently to bend the bar, not only straight, but to produce a slightly reversed curve to compensate for the tendency to spring back. If there is a reverse bend in the bar, the bar running out in opposite directions, first straighten one of the bends. The bar will then run out in one direction only, and this bend can then be straightened.

NOTE.—In case of short bends, the blocks h may be placed nearer each other. If very short bends occur near the ends, the center hole may run out of true and should be reamed again. Very large shafts, or those of high carbon steel, having short bends, should be heated before being straightened.

6. Set up the tail-center, and again test the bar with the chalk. If the bar still runs out, again move the straightener to the most pronounced bend and straighten as before. Continue in this manner until the bar is practically straight.

NOTE.—The surface of a shaft is sometimes under a tension due to the rolling process. Because of this it is at times difficult to turn a long shaft and have it perfectly straight when finished, as this tension is removed in turning the shaft, which causes it to spring and run out of true when the center-rest is removed. For this reason it is often necessary to straighten the finished shaft.

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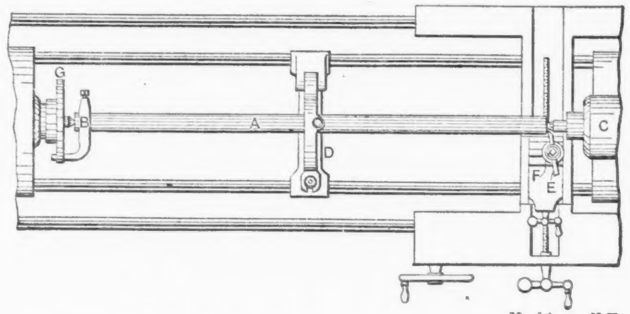
SHOP OPERATION SHEET NO. 53.

H. K. Griggs. MACHINERY, February, 1908.



To Face the Ends of a Rough Bar for Making a Shaft.

NOTE.—In the regular production of shafting, special appliances are in use by which the standard sizes are rapidly turned out. It is, however, often necessary to make shafts when these facilities are not available. In the present instance the shaft is supposed to be of such a length that a center support is necessary to prevent sagging. It is also assumed that center holes have been drilled and reamed in the ends of the rough bar.

1. Set the tail-stock C so that the lathe will take the bar A between the centers, and clamp it.

2. Fasten the lathe-dog B to one end of the bar A, and place the bar between the centers of the lathe, the dog engaging with the slot in the face-plate G. Oil the tail-center and set it up in place.

3. True up a spot, which is to be used as a bearing for the center-rest, near the middle of the bar A. This spot should not be midway between the ends of the bar, but about six inches nearer the head-stock end. When turning this spot, use a sharp pointed tool, and take light cuts, using a fine feed.

4. Clamp a center-rest D to the lathe bed, placing the center-rest so that its jaws are opposite to the spot just turned. Adjust the jaws to the work and oil their inner ends.

NOTE.—If a bar is quite long, it may be impossible to turn a spot near the middle of the bar because of its extreme flexibility; in such a case, a spot is turned on the bar as far from the dead center as possible and the center-rest jaws are adjusted to this spot. Then a second spot is turned farther along the bar, and, if necessary, the operation repeated. When it is not desirable to turn a spot for the center-rest, a "cat head" is sometimes used. This consists of a collar, about six inches long, which has four set-screws in each end. This collar fits loosely over the bar, and it is adjusted by the set-screws until it runs true. The jaws of the center-rest are then adjusted to the cylindrical surface of the cat head between the heads of the set-screws.

5. Place the facing tool E in the tool-post F, and clamp it.

6. Face the end of the bar A, feeding from the center out, by hand.

7. Take the bar out of the lathe, turn it end for end, and change the dog to the opposite end. Then face the second end as described in steps 5 and 6.

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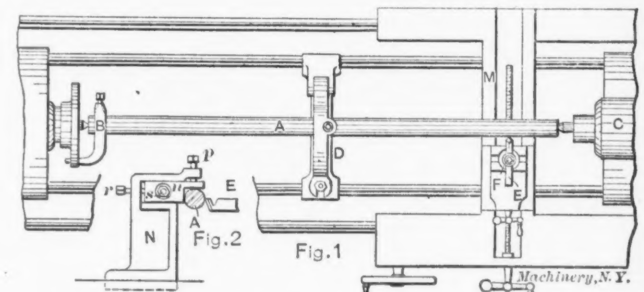
SHOP OPERATION SHEET NO. 54.

H. K. Griggs. MACHINERY, February, 1908.



To Rough Turn and Finish a Shaft from the Rough Bar.

NOTE.—The bar A is supposed to have been center-drilled and reamed; to have been straightened; to have had the ends faced and a spot turned near the middle for a center-rest bearing.

1. Adjust the jaws of the center-rest D to the turned spot on the bar A. Care should be taken to have the jaws bear evenly on the bar to avoid springing it, and when the jaws are adjusted, the bar should rotate easily. Oil the inner ends of the jaws.

2. In the tool-post F, clamp a diamond point roughing tool E and turn 3 or 4 inches of the end of the bar to within 1/32 inch of the finish diameter.

3. Upon the rear of the carriage M, fix the follow-rest N (Fig. 2) with its jaw n in close contact with the turned portion. Adjust the jaw n by the set-screws p and r, and clamp it by the bolt s. This follow-rest steadies the bar when it is being turned, and the roughing cut may now be continued until the carriage is prevented from going farther by the center-rest D.

4. Turn the bar A end for end, and change the dog B to the opposite end. Rough turn this half of the bar as described in steps 2 and 3.

NOTE.—When rough turning a shaft, it will become more or less heated which will cause elongation. Because of this it is necessary to slacken the tail-center at times, especially when turning long shafts, to avoid springing them.

5. After taking the roughing cut, unscrew the center-rest jaws, speed up the lathe and see if the rough-turned shaft runs true. If it does not run true, again turn a spot for the center-rest jaws.

6. Again adjust the jaws of the center-rest to the work, and replace the roughing tool E with a finishing tool having a straight cutting edge and slightly rounded corners. Set the tool to the exact finish diameter, and take the finishing cuts in the manner described for the roughing cuts in steps 3 and 4. When taking this cut the tool should be lubricated with soda water, or some soapy mixture. A lubricant of this kind can also be used to advantage when taking the roughing cuts.

NOTE.—Regular shafting lathes are sometimes arranged so that a shaft can be driven from either end. This is desirable when turning a long shaft, as otherwise the torsional stress at the beginning of the cut would be considerable.

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springing them. Bring down the pre[illegible]ock *j* upon the bar *A*, and with the lever *k* turn the screw *g* sufficiently to bend the bar, not only straight, but to produce a slightly reversed curve to compensate for the tendency to spring back. If there is a reverse bend in the bar, the bar running out in opposite directions, first straighten one of the bends. The bar will then run out in one direction only, and this bend can then be straightened.

NOTE.—In case of short bends, the blocks *h* may be placed nearer each other. If very short bends occur near the ends, the center hole may run out of true and should be reamed again. Very large shafts, or those of high carbon steel, having short bends, should be heated before being straightened.

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NOTE.—The surface of a shaft is sometimes under a tension due to the rolling process. Because of this it is at times difficult to turn a long shaft and have it perfectly straight when finished, as this tension is removed in turning the shaft, which causes it to spring and run out of true when the center-rest is removed. For this reason it is often necessary to straighten the finished shaft.

[illegible] and take light [illegible]

4. Clamp a center-rest *D* to the lathe bed, placing the center-rest so that its jaws are opposite to the spot just turned. Adjust the jaws to the work and oil their inner ends.

NOTE.—If a bar is quite long, it may be impossible to turn a spot near the middle of the bar because of its extreme flexibility; in such a case, a spot is turned on the bar as far from the dead center as possible and the center-rest jaws are adjusted to this spot. Then a second spot is turned farther along the bar, and, if necessary, the operation repeated. When it is not desirable to turn a spot for the center-rest, a "cat head" is sometimes used. This consists of a collar, about six inches long, which has four set-screws in each end. This collar fits loosely over the bar, and it is adjusted by the set-screws until it runs true. The jaws of the center-rest are then adjusted to the cylindrical surface of the cat head between the heads of the set-screws.

5. Place the facing tool *E* in the tool-post *F*, and clamp it.

6. Face the end of the bar *A*, feeding from the center out, by hand.

7. Take the bar out of the lathe, turn it end for end, and change the dog to the opposite end. Then face the second end as described in steps 5 and 6.

[illegible] in steps 2 and 3.

NOTE.—When rough turning a shaft, it will become more or less heated which will cause elongation. Because of this it is necessary to slacken the tail-center at times, especially when turning long shafts, to avoid springing them.

5. After taking the roughing cut, unscrew the center-rest jaws, speed up the lathe and see if the rough-turned shaft runs true. If it does not run true, again turn a spot for the center-rest jaws.

6. Again adjust the jaws of the center-rest to the work, and replace the roughing tool *E* with a finishing tool having a straight cutting edge and slightly rounded corners. Set the tool to the exact finish diameter, and take the finishing cuts in the manner described for the roughing cuts in steps 3 and 4. When taking this cut the tool should be lubricated with soda water, or some soapy mixture. A lubricant of this kind can also be used to advantage when taking the roughing cuts.

NOTE.—Regular shafting lathes are sometimes arranged so that a shaft can be driven from either end. This is desirable when turning a long shaft, as otherwise the torsional stress at the beginning of the cut would be considerable.

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SOLUTION OF OBLIQUE ANGLE TRIANGLES.

Parts Given	Parts to be found: $a=$	$b=$	$c=$	$\angle A$	$\angle B$	$\angle C$
$a-b-c$				$\frac{b^2+c^2-a^2}{2bc}=\cos A$	$\frac{a^2+c^2-b^2}{2ac}=\cos B$	$\frac{a^2+b^2-c^2}{2ab}=\cos C$
$b-c-\angle A$	$\sqrt{b^2+c^2-2bc\cos A}$				$\frac{b\sin A}{c-b\cos A}=\tan B$	$\frac{c\sin A}{b-c\cos A}=\tan C$
$a-c-\angle B$		$\sqrt{a^2+c^2-2ac\cos B}$		$\frac{a\sin B}{c-a\cos B}=\tan A$		$\frac{c\sin B}{a-c\cos B}=\tan C$
$a-b-\angle C$			$\sqrt{a^2+b^2-2ab\cos C}$	$\frac{a\sin C}{b-a\cos C}=\tan A$	$\frac{b\sin C}{a-b\cos C}=\tan B$	
$a-b-\angle A$			$\frac{a\sin C}{\sin A}$		$\frac{b\sin A}{a}=\sin B$	$180^\circ-(A+B)$

Contributed by E. A. Johnson.

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MACHINERY, February, 1908.
Data Sheet No. 84.

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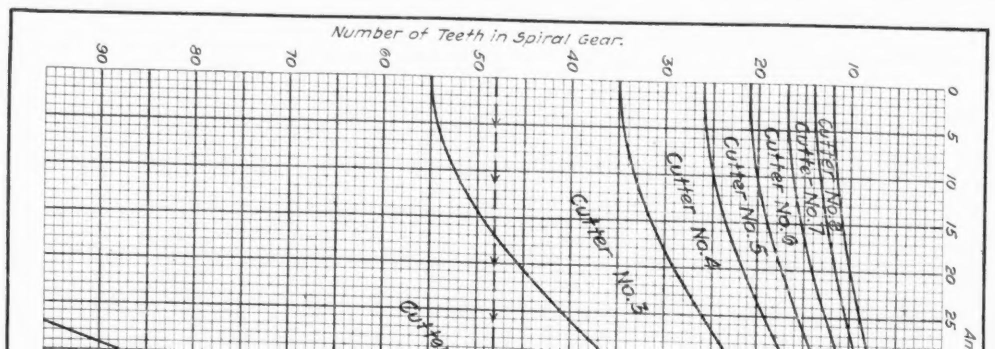
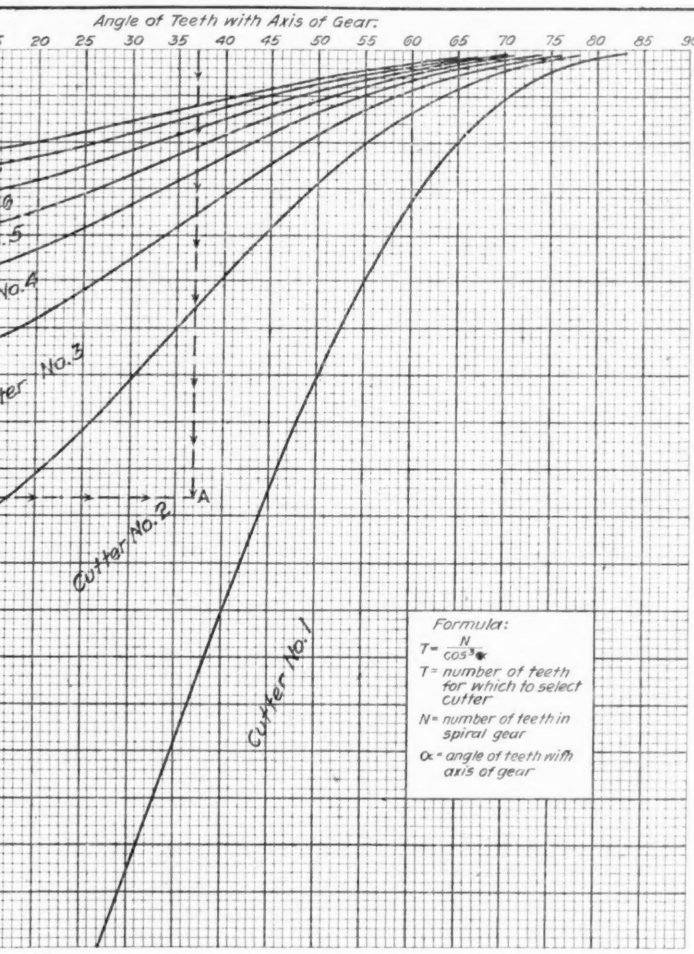


Contributed by Elmer G. Eberhardt.

MACHINERY'S DATA SHEETS.

Contributions and data suitable for publication in these Data Sheets are solicited. Payment will be made for all accepted matter.

DIAGRAM FOR FINDING SPIRAL GEAR CUTTER NUMBERS.



The cutter used in milling spiral gears is a standard spur gear cutter, the only difference being that the number of the cutter used to cut a spiral gear is not necessarily the same as that used to cut a spur gear of the same number of teeth. The angle of the teeth of a spiral gear with its axis affects the tooth form, and therefore, the number of the cutter.

The selection of the cutter is fixed by the formula given in the lower right-hand corner of the diagram. The delimiting curves thereon were plotted by the formula, the area between the curves being the field of intersection of the combinations of angles and numbers of teeth covered by each designated cutter number.

For example, suppose the angle of the teeth of a gear is 37 degrees with its axis, and the number of teeth is 48. The point A, at which the horizontal line (representing the tooth number), and the vertical line (representing the angle) intersect, falls within the area marked "Cutter No. 2" Therefore, a No. 2 cutter is required to cut a 48-tooth spiral gear having the teeth at an angle of 37 degrees with its axis.

. Eberhardt.

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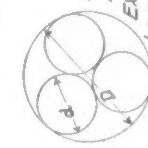
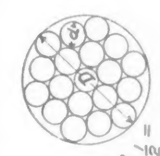
TABLE OF INSCRIBED TANGENT CIRCLES.

Approximate Formulas:

$N = 0.907\left(\frac{D}{d} - 0.94\right)^2 + 3.7. \quad R = 0.94 + \sqrt{\frac{N-3.7}{0.907}}.$



N = Number of inscribed circles.
R = Diameter of large circle ÷ diameter of small circle.
Example:- How many wires $\frac{1}{2}$ inch in diameter can be placed inside a 5 inch pipe? The value of R is $5 \div \frac{1}{2}$ = 10, and looking for this number in table, find by interpolation N = 76.



N	R	N	R	N	R	N	R	N	R
2	2.00	34	6.76	130	12.80	290	18.75	600	26.65
3	2.15	35	6.86	135	13.06	295	18.90	610	26.86
4	2.41	36	[illegible]	140	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

Contributed by E H. Lockwood.

a-c-∠B		$\sqrt{a^2+c^2-2ac\cos B}$		$\frac{a\sin B}{c-a\cos B}=\tan A$		$\frac{c\sin B}{a-c\cos B}=\tan C$
a-b-∠C			$\sqrt{a^2+b^2-2ab\cos C}$	$\frac{a\sin C}{b-a\cos C}=\tan A$	$\frac{b\sin C}{a-b\cos C}=\tan B$	
a-b-∠A			$\frac{a\sin C}{\sin A}$		$\frac{b\sin A}{a}=\sin B$	180°−(A+B)
a-b-∠B			$\frac{b\sin C}{\sin B}$	$\frac{a\sin B}{b}=\sin A$		180°−(A+B)
a-c-∠A		$\frac{a\sin B}{\sin A}$			180°−(A+C)	$\frac{c\sin A}{a}=\sin C$
a-c-∠C		$\frac{c\sin B}{\sin C}$		$\frac{a\sin C}{c}=\sin A$	180°−(A+C)	
b-c-∠B	$\frac{b\sin A}{\sin B}$			180°−(B+C)		$\frac{c\sin B}{b}=\sin C$
b-c-∠C	$\frac{c\sin A}{\sin C}$			180°−(B+C)	$\frac{b\sin C}{c}=\sin B$	
a-∠A-∠B		$\frac{a\sin B}{\sin A}$	$\frac{a\sin C}{\sin A}$			180°−(A+B)
a-∠A-∠C		$\frac{a\sin B}{\sin A}$	$\frac{a\sin C}{\sin A}$		180°−(A+C)	
a-∠B-∠C		$\frac{a\sin B}{\sin A}$	$\frac{a\sin C}{\sin A}$	180°−(B+C)		
b-∠A-∠B	$\frac{b\sin A}{\sin B}$		$\frac{b\sin C}{\sin B}$			180°−(A+B)
b-∠A-∠C	$\frac{b\sin A}{\sin B}$		$\frac{b\sin C}{\sin B}$		180°−(A+C)	
b-∠B-∠C	$\frac{b\sin A}{\sin B}$		$\frac{b\sin C}{\sin B}$	180°−(B+C)		
c-∠A-∠B	$\frac{c\sin A}{\sin C}$	$\frac{c\sin B}{\sin C}$				180°−(A+B)
c-∠A-∠C	$\frac{c\sin A}{\sin C}$	$\frac{c\sin B}{\sin C}$			180°−(A+C)	
c-∠B-∠C	$\frac{c\sin A}{\sin C}$	$\frac{c\sin B}{\sin C}$		180°−(B+C)		

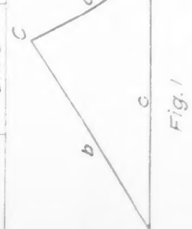


Fig. 1

Note 1:- By means of the table any part of an oblique triangle may be found when any three other parts are given, with the following exception:

Given two sides and the angle opposite one of them; then, if the side opposite is less than the adjacent × the sine of the angle, the triangle is impossible; or if the side opposite = the adjacent × the sine of the angle, the triangle is a right triangle; or if the side opposite is less than the adjacent but does not come under the above, the triangle is capable of two solutions and can be drawn as in Fig. 2 as well as in Fig. 1.

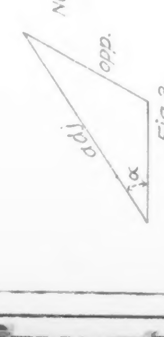


Fig. 2.

Note 2:- In some cases two steps are necessary to solve, as for example, having given sides a and b and angle A, to find c: The formula reads $c = \frac{a\sin C}{\sin A}$ but angle C must first be derived from C = 180° − (A + B), and the same applies to other angles in certain cases as is apparent above.

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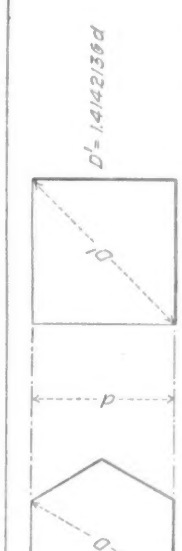
SQUARE AND HEXAGON NUT DIAMETERS.



d	D	D'	d	D	D'	d	D	D'
$\frac{1}{4}$	0.2886	0.3535	$1\frac{1}{4}$	1.4434	1.7677	$2\frac{5}{16}$	2.6702	3.2703
$\frac{9}{32}$	0.3247	0.3977	$1\frac{9}{32}$	1.4794	1.8119	$2\frac{3}{8}$	2.7424	3.3587
$\frac{5}{16}$	0.3608	0.4419	$1\frac{5}{16}$	1.5155	1.8561	$2\frac{7}{16}$	2.8145	3.4471
$\frac{11}{32}$	0.3968	0.4861	$1\frac{11}{32}$	1.5516	1.9003	$2\frac{1}{2}$	2.8867	3.5355

Contributed by Howard D. Yoder.

...BERS.

...er used in milling spiral gears is ...spur gear cutter, the only difference ...the number of the cutter used ...ral gear is not necessarily the ...hat used to cut a spur gear of the ...ber of teeth. The angle of the ...piral gear with its axis affects ...rm, and therefore, the number ...er.

...ction of the cutter is fixed by the ...ven in the lower right-hand cor- ...diagram. The delimiting curves ...re plotted by the formula, the ...en the curves being the field of ...n of the combinations of angles ...rs of teeth covered by each ...d cutter number.

...mple, suppose the angle of the ...ear is 37 degrees with its axis, ...mber of teeth is 48. The point ... the horizontal line (represent- ...th number), and the vertical line ...ng the angle) intersect, falls ...area marked "Cutter No. 2" ...a No. 2 cutter is required to cut ... spiral gear having the teeth at ...f 37 degrees with its axis.

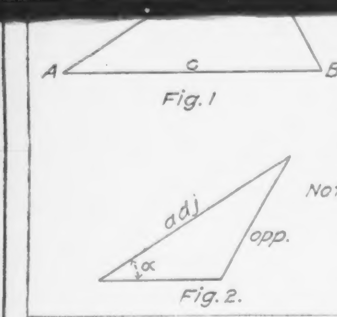


Fig. 1

Fig. 2.

...the side opposite is less than the adjacent x the sine of the angle, the triangle is impossible; or if the side opposite = the adjacent x the sine of the angle, the triangle is a right triangle; or if the side opposite is less than the adjacent but does not come under the above, the triangle is capable of two solutions and can be drawn as in Fig. 2 as well as in Fig. 1.

Note 2:- In some cases two steps are necessary to solve, as for example, having given sides a and b and angle A, to find c: The formula reads $c = \frac{a \sin C}{\sin A}$ but angle C must first be derived from $C = 180° - (A+B)$, and the same applies to other angles in certain cases as is apparent above.

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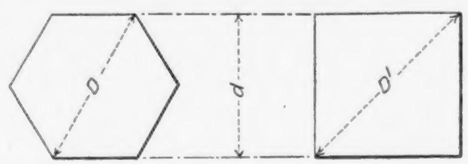
SQUARE AND HEXAGON NUT DIAMETERS.

$D = 1.154694d$

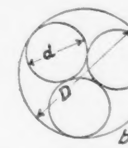


$D' = 1.4142136d$

d	D	D'	d	D	D'	d	D	D'
1/4	0.2886	0.3535	1 1/4	1.4434	1.7677	2 5/16	2.6702	3.2703
9/32	0.3247	0.3977	1 9/32	1.4794	1.8119	2 3/8	2.7424	3.3587
5/16	0.3608	0.4419	1 5/16	1.5155	1.8561	2 7/16	2.8145	3.4471
11/32	0.3968	0.4861	1 11/32	1.5516	1.9003	2 1/2	2.8867	3.5355
3/8	0.4329	0.5303	1 3/8	1.5877	1.9445	2 9/16	2.9589	3.6239
13/32	0.4690	0.5745	1 13/32	1.6238	1.9887	2 5/8	3.0311	3.7123
7/16	0.5051	0.6187	1 7/16	1.6598	2.0329	2 11/16	3.1032	3.8007
15/32	0.5412	0.6629	1 15/32	1.6959	2.0771	2 3/4	3.1754	3.8891
1/2	0.5773	0.7071	1 1/2	1.7320	2.1213	2 13/16	3.2476	3.9794
17/32	0.6133	0.7513	1 17/32	1.7681	2.1655	2 7/8	3.3197	4.0658
9/16	0.6494	0.7955	1 9/16	1.8042	2.2097	2 15/16	3.3919	4.1542
19/32	0.6855	0.8397	1 19/32	1.8403	2.2539	3	3.4641	4.2426
5/8	0.7216	0.8839	1 5/8	1.8764	2.2981	3 1/16	3.5362	4.3310
21/32	0.7576	0.9281	1 21/32	1.9124	2.3423	3 1/8	3.6084	4.4194
11/16	0.7937	0.9723	1 11/16	1.9485	2.3865	3 3/16	3.6806	4.5078
23/32	0.8298	1.0164	1 23/32	1.9846	2.4306	3 1/4	3.7527	4.5962
3/4	0.8659	1.0606	1 3/4	2.0207	2.4708	3 5/16	3.8249	4.6846
25/32	0.9020	1.1048	1 25/32	2.0568	2.5190	3 3/8	3.8971	4.7729
13/16	0.9380	1.1490	1 13/16	2.0929	2.5632	3 7/16	3.9692	4.8613
27/32	0.9741	1.1932	1 27/32	2.1289	2.6074	3 1/2	4.0414	4.9497
7/8	1.0102	1.2374	1 7/8	2.1650	2.6516	3 9/16	4.1136	5.0381
29/32	1.0463	1.2816	1 29/32	2.2011	2.6958	3 5/8	4.1857	5.1265
15/16	1.0824	1.3258	1 15/16	2.2372	2.7400	3 11/16	4.2579	5.2149

Contributed by Howard D. Yoder.

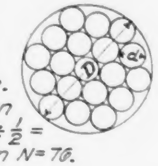
TABLE OF INSCRIBED TANGENT CIRCLES.



Approximate Formulas:

$N = 0.907\left(\frac{D}{d} - 0.94\right)^2 + 3.7. \qquad R = 0.94 + \sqrt{\frac{N-3.7}{0.907}}.$

N = Number of inscribed circles.
R = Diameter of large circle ÷ diameter of small circle.
Example:- How many wires 1/2 inch in diameter can be placed inside a 5 inch pipe? The value of R is 5 ÷ 1/2 = 10, and looking for this number in table, find by interpolation N = 76.



N	R	N	R	N	R	N	R	N	R
2	2.00	34	6.76	130	12.80	290	18.75	600	26.65
3	2.15	35	6.86	135	13.06	295	18.90	610	26.86
4	2.41	36	7.00	140	13.26	300	19.05	620	27.07
5	2.70	37	7.00	145	13.49	310	19.35	630	27.28
6	3.00	38	7.08	150	13.72	320	19.65	640	27.49
7	3.00	39	7.18	155	13.95	330	19.94	650	27.70
8	3.31	40	7.31	160	14.17	340	20.23	660	27.91
9	3.61	41	7.39	165	14.39	350	20.52	670	28.12
10	3.80	42	7.43	170	14.60	360	20.81	680	28.33
11	3.92	43	7.61	175	14.81	370	21.09	690	28.54
12	4.05	44	7.70	180	15.01	380	21.36	700	28.75
13	4.23	45	7.72	185	15.20	390	21.63	720	29.14
14	4.41	46	7.81	190	15.39	400	21.90	740	29.52
15	4.55	47	7.92	195	15.57	410	22.17	760	29.90
16	4.70	48	8.00	200	15.75	420	22.44	780	30.28
17	4.86	49	8.03	205	15.93	430	22.70	800	30.65
18	5.00	50	8.13	210	16.11	440	22.96	820	31.02
19	5.00	55	8.21	215	16.29	450	23.21	840	31.39
20	5.18	60	8.94	220	16.46	460	23.47	860	31.75
21	5.31	65	9.25	225	16.63	470	23.72	880	32.11
22	5.49	70	9.61	230	16.80	480	23.97	900	32.46
23	5.61	75	9.93	235	16.97	490	24.21	920	32.80
24	5.72	80	10.20	240	17.14	500	24.45	940	33.14

Contributed by E. H. Lockwood.

be readily bound. It may be cut into four
nd into note-book form for convenient re-
ed into holes punched at points indicated.

21	5.31	65	9.25	225	16.63	470	23.72	880	32.11
22	5.49	70	9.61	230	16.80	480	23.97	900	32.46
23	5.61	75	9.93	235	16.97	490	24.21	920	32.80
24	5.72	80	10.20	240	17.14	500	24.45	940	33.14
25	5.81	85	10.46	245	17.30	510	24.68	960	33.48
26	5.92	90	10.73	250	17.46	520	24.91	980	33.82
27	6.00	95	11.15	255	17.63	530	25.13	1000	34.15
28	6.13	100	11.34	260	17.79	540	25.35	1100	35.75
29	6.23	105	11.60	265	17.95	550	25.57	1200	37.30
30	6.40	110	11.85	270	18.11	560	25.79	1300	38.80
31	6.44	115	12.10	275	18.27	570	26.01	1400	40.20
32	6.55	120	12.34	280	18.43	580	26.23	1500	41.60
33	6.70	125	12.67	285	18.59	590	26.44	1600	42.95

27/32	0.9741	1.1932	1 21/32	2.1289	2.6074	3 1/2	4.0414	4.9497
7/8	1.0102	1.2374	1 7/8	2.1650	2.6516	3 9/16	4.1136	5.0381
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31/32	1.1184	1.3700	1 31/32	2.2733	2.7842	3 3/4	4.3301	5.3033
1	1.1547	1.4142	2	2.3094	2.8284	3 13/16	4.4023	5.3917
1 1/32	1.1907	1.4584	2 1/32	2.3453	2.8726	3 7/8	4.4744	5.4801
1 1/16	1.2268	1.5026	2 1/16	2.3815	2.9168	3 15/16	4.5466	5.5684
1 3/32	1.2629	1.5468	2 3/32	2.4176	2.9610	4	4.6188	5.6568
1 1/8	1.2990	1.5910	2 1/8	2.4537	3.0052	4 1/8	4.7631	5.8336
1 5/32	1.3351	1.6352	2 5/32	2.4898	3.0494	4 1/4	4.9047	6.0104
1 3/16	1.3712	1.6793	2 3/16	2.5259	3.0936	4 3/8	5.0518	6.1872
1 7/32	1.4073	1.7235	2 1/4	2.5981	3.1820	4 1/2	5.1961	6.3639

Superheated Vapors; The Steam Engine; Compound Engines; Testing Steam Engines; Influence of the Cylinder Walls; Economy of Steam Engines; Friction of Engines; Internal Combustion Engines; Compressed Air; Refrigerating Machines; Flow of Fluids; Injectors; Steam Turbines.

CATALOGUES AND CIRCULARS.

T. B. Wood's Sons Co., Chambersburg, Pa. Catalogue No. 23, illustrating friction clutches, pulleys, etc.

Ferracute Machine Co., Bridgeton, N. J. Circular illustrating some of the many sizes and styles of Ferracute presses.

United Engineering & Foundry Co., Pittsburg, Pa. Catalogue of roll turning lathes made in 16-, 18-, 26-, 34-, 44- and 60-inch sizes.

United Engineering & Foundry Co., Pittsburg, Pa. Catalogue of high-speed saws for iron and steel works, including both hot and cold saws.

Buckeye Engine Co., Salem, O. Folder entitled "An Illustrated Talk on Halos," being an advertisement of the Buckeye electric blue-printing machine.

Goldschmidt Thermit Co., 90 West St., New York. Pamphlet on the Thermit Welding Process, illustrated with many examples of work done.

R. Woodman Mfg. & Supply Co., 63 Oliver St., Boston, Mass. Catalogue of railroad ticket punches, baggage checks and hat badges. A list of 444 dies, with illustrations of shapes cut out, is included.

Fosdick Machine Tool Co., Cincinnati, Ohio. Specification circular of the Fosdick No. 2 (Style D) horizontal boring, drilling and milling machine, which was illustrated and described in our September, 1907, issue.

Frank Mossberg Co., Attleboro, Mass. Catalogue No. 16, listing drop forged screw wrenches of the bicycle type, bicycle bells, advertising novelties such as paper cutters, etc., punches and dies, special machinery and screw machine products.

B. F. Sturtevant Co., Hyde Park, Mass. Bulletin No. 151, describing the Sturtevant steam turbine. The bulletin gives an interesting historical sketch of the steam turbine as a preface to the general matter. The Sturtevant machine is illustrated in detail, the details including the nozzles, bearings, governor, bucket wheels, vanes, etc.

General Electric Co., Schenectady, N. Y. Bulletin No. 4546, "The Electrification of the West Shore Railroad." The electrification of the West Shore R. R. between Utica and Syracuse is described, including the 60,000-volt transmission line, the inverted third-rail and the hydro-electric power development.

Fox Machine Co., 815-825 No. Front St., Grand Rapids, Mich. Catalogue No. 29 of Fox light milling machines, made in three sizes. This machine was developed to meet the requirements of typewriter manufacture, and is well calculated to meet the needs of manufacture of all light machine parts.

Mitts & Merrill, 843 Water St., Saginaw, Mich. New catalogue entitled The Giant Key-Seater, describing this machine and its parts in detail, and illustrating various other types of key-seating machines manufactured by this company. A section of the book is also devoted to key-seat milling machines.

Fox Machine Co., 815-825 No. Front St., Grand Rapids, Mich. Catalogue No. 67 of Fox tube and pipe cutters. These machines are of the rotary cutter type largely used in railway shops for cutting off boiler flues. They do not include threading attachments, but are confined to cutting-off work alone. The machines are listed in four sizes.

Fox Machine Co., 815-825 No. Front St., Grand Rapids, Mich. Catalogue No. 30 of Fox high-speed sensitive drilling machines made with one, two, three, four, five and six spindles. Each spindle has three changes of speed and independent belt tension adjustment. Other features of distinction are listed, including hardened and ground upper cone shafts, etc.

Kilbourne & Jacobs Mfg. Co., Columbus, Ohio. Circular of steel factory equipment, such as pressed steel pans, tote boxes, steel barrels, barrel trucks, etc. The pressed steel pans are made from one piece of heavy sheet steel. Having rounded corners and no rough surfaces whatever, they are well adapted for handling fine machine shop products without injury.

General Electric Co., Schenectady, N. Y. Bulletin No. 4537, "The Electric Locomotive in Heavy Passenger and Freight Work," in which is described a large number of present and proposed types of electric locomotives. Sketches are given of locomotives ranging in weight from 17 to 150 tons and adapted to mining, high-speed passenger, slow-speed freight, mountain grade trunk lines, etc. Valuable data are included.

General Electric Co., Schenectady, N. Y. Booklet giving standard symbols for wiring plans, adopted by the National Electric Contractors' Association and the American Institute of Architects, in a form that illustrates their use clearly. Sample plans of an office, kitchen and dining-room are shown with the electric wiring symbols to indicate the various appliances. The booklet also illustrates the General Electric specialties used in house and office electric installations.

Chandler Planer Co., Ayer, Mass. Bulletins M-3, M-4, M-5, M-6, M-7 and M-9 descriptive of the Chandler planers. Bulletin M-3 contains a bit of history of planer development and enlarges on the special features of the Chandler design, among which are case-hardened shafts, feed-regulating mechanism, change-speed mechanism, and finally and chief, high speed. Cutting speed is put at 30 feet for frog and switch planers, and ranges up to 90 feet for brass-working planers; the return stroke is made at speeds ranging from 100 to 150 or 200 feet per minute.

Cling Surface Co., 1018 Niagara St., Buffalo, N. Y. Booklet entitled "The Treatment of Belts and Ropes for Service and Profit," which gives the best methods of applying "Cling Surface" to leather, canvas, cotton, balata, link, camel-hair, rubber and rope belts. It also gives many valuable hints on the general treatment of belts. Illustrations of several typical belt drives on which "Cling Surface" is used, are included along with details of certain interesting tests. The booklet contains 87 pages and will be found of interest and value by anyone having the care of belting.

Hammacher, Schlemmer & Co., 4th Ave. and 13th St., New York. Circular descriptive of a portable machinist's tool chest which appears to be what many journeymen need. Its outside measurements are: Length, 15¾ inches; height, 10¾ inches; width, 8 inches. The drawers are felt-lined and run on hard wood slides. The front is provided with two hinged joints which permit it to fold back neatly and expose the drawers, of which there are three full length (14 inches) and five half length (6¾ inches). A flexible handle in the center of the top permits the chest to be carried the same as a suit-case.

Hess-Bright Mfg. Co., Philadelphia, Pa. Sheet, 9 x 12 inches, showing Hess-Bright ball bearing mounted for radial load without thrust. This sheet is the first of series entitled "Ball Bearings and Their Correct Use," which will show the principles of correct ball bearing mounting and scale drawings of various mechanisms in which ball bearings are used. The new series is an extension of the 3 x 5-inch index data card series issued by the company, which has met with much favor, but which, for the present purpose, is too small. The sheets are punched for a binder, thus permitting them to be gathered together in convenient form for reference.

Newton Machine Tool Works, Inc., Philadelphia, Pa. General catalogue No. 45 of machine tools, containing 303 pages and illustrating a great variety of special and standard machine tools. These include boring machines for bridge chords, cylinders, locomotive rods, motor frames and many other purposes; cold saw cutting-off machines; crank planers; drilling machines; gear cutters; milling machines; rotary planing machines; shaping machines; slotters; and miscellaneous tools not readily classifiable. The catalogue illustrates a number of the special floor-plate slotters, boring machines and drilling machines in use in the General Electric Works, Schenectady, N. Y., in which shops portable tool practice on floor-plates has reached a high development.

MANUFACTURERS' NOTES.

S. J. Smith Machinery Co. and L. Booth & Sons, Los Angeles, Cal., have consolidated as the Smith-Booth-Usher Co., with general offices and salesrooms at 212-214 South Los Angeles St.

Independent Pneumatic Tool Co. has moved its San Francisco office and storeroom from 11 Front St. to much larger quarters at 61 Fremont St., in the heart of the machinery district.

Rockford Drilling Machine Co., Rockford, Ill., is the successor to B. F. Barnes Co., of same city. The change is in the name of the company only, the place of business and product remaining the same as heretofore.

Wile Power Gas Co., Cleveland, Ohio, manufacturer of producer gas installations for power and fuel, has removed its general offices from the Cutler Bldg., Rochester, N. Y., to its works, 1688-92 Columbus Road, Cleveland, Ohio.

Gardner Machine Co., which has its main office and factory in Beloit, Wis., has opened a branch office in Cleveland, Ohio, 505 Caxton Building. The branch office will enable the company to take care of its increasing Central and Eastern business.

American Museum of Safety Devices and Industrial Hygiene, 231 W. 39th St., New York, announces that a friend of the Museum has offered a prize of $100 for the best essay on the "Economic Waste of Accidents." The committee of award consists of Richard Watson Gilder, George Gilmour and W. H. Tolman.

Coes Wrench Co., Worcester, Mass., announces that owing to the numerous requests for its 1908 souvenir, "A Vestal Virgin," by Sichel, it has had a limited number printed in Germany in full colors without calendar attached, size 20 x 29 inches, which will be sent, one only to an individual, on receipt of 40 cents to cover cost of wrapping and postage.

The name of the concern mentioned in the "Manufacturers' Notes" appearing in December as being the builder of the new Libbey turret lathe should have been the International Machine Tool Co. instead of the Indianapolis Machine Tool Co., as it erroneously appeared. The company is erecting a new factory and expects to be in full operation this month.

Peter A. Frasse & Co., New York, have opened a branch warehouse at 408 Commerce St., Philadelphia, Pa., for the sale of Shelby steel tube, Poldi tool steels, drill rod, music wire, Poldi high-speed drills, etc. This branch warehouse will handle only such specialties as the above and will not carry the full line of supplies to be had at their New York store, 94 Fulton St.

Hartford Blower Co., Hartford, Conn., recently installed a very large dust collecting system for Comstock, Cheney & Co., Ivorytin, Conn., for handling shavings, saw-dust, etc. The total height of the collector is 35 feet 5 inches, and the weight of the material used was 13,287 pounds. There are three discharge pipes, two being 40 inches diameter and the third 52 inches diameter. This installation is the fourth that the Hartford Blower Co. has made for this concern.

A. H. Briggs, manager of the New York store of the L. S. Starrett Co., Athol, Mass., since its opening ten years ago, has resigned to go into business for himself. He will be associated with Mr. W. M. Briggs, New York manager for the Rock Island Tool Co., Rock Island, Ill., manufacturers of vises, and the Joyce Cridland Co., Dayton, Ohio, makers of jacks. The firm will represent manufacturers of first-class lines of tools, etc.

S. Obermayer Co., Cincinnati, Ohio, is sending two slips with all letters, entitled "Put Your Shoulder to the Wheel—NOW," and "To the Man who Signs the Checks," entreating all to pay their bills promptly, and asking the recipients of checks from the S. Obermayer Co. to keep the ball rolling. In brief, the slips make a sensible and patriotic appeal to stop hoarding and to help business by paying all bills promptly.

G. M. Yost Mfg. Co., Mechanicsburg, Pa., is building a new factory at Meadville, Pa., to which place the company will move both the manufacturing end and business offices about March 1. About $5,000 worth of new machinery will be added to the equipment. The new plant will be organized and equipped for the manufacture of vises of all kinds and for all purposes, and will be one of the best plants in America. The business depression has not affected the company's business to an appreciable extent so far, and it regards the future in the most optimistic spirit.

Pittsburg Automatic Vise and Tool Co., Pittsburg, Pa., has recently patented an improvement, which it is applying to the jaws of its line of vises. This improvement consists simply in cutting a half-round groove of a radius of about ⅝ inch, lengthwise of the face of one of the jaws of the vise. The purpose of this groove is to permit the holding of round bar stock, tubes, etc., with a much greater degree of firmness and precision than is possible with flat gripping surfaces. In wood-working vises, the groove is of considerably larger diameter to agree with the class of work for which it is intended.

Independent Pneumatic Tool Co., Chicago, Ill., held a meeting of the stockholders in Jersey City, N. J., recently, at which the officers were re-elected, as follows: James B. Brady, president; W. O. Jacquette, first vice-president; John D. Hurley, second vice-president; A. B. Holmes, secretary and treasurer. The annual statement shows a large increase of business over the previous year. The company has greatly enlarged its Aurora, Ill., plant, and is in good financial condition. It is optimistic as regards future business. No cancellations of orders for "Thor" pneumatic tools have been received, nor has the company cancelled any of its orders for materials or machinery.

Curtis & Curtis Co., 8 Garden St., Bridgeport, Conn., which for the past twenty-five years has been engaged in the manufacture of pipe cutting and threading machinery, has just completed a large addition to its plant. The roof of one of the main buildings was raised, thus providing 8,000 square feet additional space for the machine shop and offices. The increased space was badly needed, the company for some time past having found it difficult to meet the demands for the Forbes die stocks, etc. The increased space will enable the manufacture of this and other specialties to be carried on in a more effective and economical manner. The increased office space has been fitted up correspondingly to properly care for increased business.

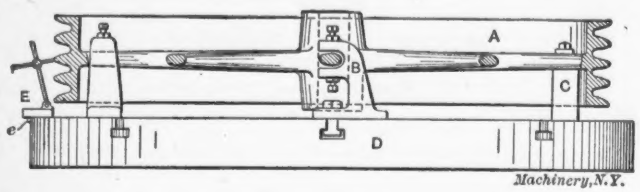
American Museum of Safety Devices and Industrial Hygiene, 231 W. 39th St., New York, announces that Prof. F. R. Hutton, past president of the American Society of Mechanical Engineers, is the chairman of the committee on admission of exhibits for the Museum. The Museum desires exhibits of devices and processes of safeguarding life and limb in connection with woodworking machinery, railway and marine transportation, mining, agriculture and manufacturing of all

CUT OUT ON THIS LINE.

PUNCH PUNCH

SHOP OPERATION SHEET NO. 49.

W. L. McLaren. MACHINERY, January, 1908.



To Set Up a Sheave Pulley Casting for Boring and Turning on a Vertical Boring Mill.

1. Select as many clamp blocks *B* as there are arms to the pulley casting. These clamp blocks should be high enough to raise the casting *A* above the table *D* far enough to allow facing the under side of the rim. Measure the distance from the center of the casting *A* to the inside of the rim, and then fasten the clamp blocks *B* to the mill table, making the distance from the outside of each block to the center of the table about ½ inch less than this measurement.

2. Place the pulley casting *A* on the boring mill table *D*, and insert the arms of the pulley into the slots in the clamp blocks *B*. Set the pulley approximately central with the table by shifting the pulley until the distances between the inside of its rim and the clamp blocks are about equal.

3. Set the rim of the pulley parallel with the mill table. This can be done by measuring from the table to various points around the rim, using a surface gage. Adjust the pulley with the lower set-screws in the clamp blocks *B*.

4. Now place the surface gage *E* upon the table in the position shown. Lower the rear pins *e* of the gage, and hold them firmly against the mill table. Set the pointer of the gage about ¼ inch from the pulley casting, and then move the gage to various points around the table, until the place where the casting is closest to the pointer is found. Set the pointer against the casting at this place, then move the gage to the opposite side of the table and measure the distance between the pointer and the casting. Move the casting toward the pointer a distance equal to one-half this measurement. Again test the casting for height, and if necessary, change lower set-screws in blocks *B* until the rim is again parallel with the table. Tighten the upper set-screws lightly.

5. Start the machine and hold a piece of chalk against the outside of the rim. If the casting is not central with the table a mark will be made on that part of the rim which is farthest from the center. Adjust the casting if necessary, by hitting it lightly with a soft hammer, until it runs perfectly true.

NOTE.—If clamp blocks *B* cannot be obtained, support the arms by blocks *C*, and adjust the height of the casting by placing thin pieces of iron or brass between the blocks and the arms. Clamp the arms to the table by using bolts and straps. If the blocks *C* are used, it will be necessary to have a block or angle-piece bolted to the boring mill table, and against an arm, to act as a driver.

PUNCH PUNCH

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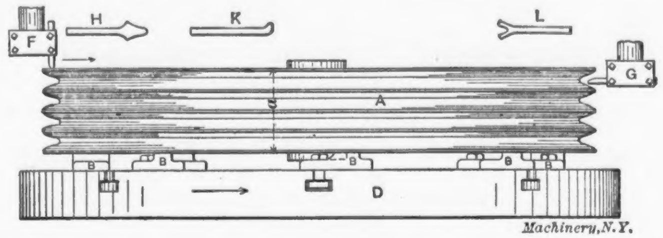
SHOP OPERATION SHEET NO. 50.

W. L. McLaren. MACHINERY, January, 1908.



To Turn and Face a Sheave Pulley Casting on a Vertical Boring Mill.

1. Set the cross-rail of the machine about six inches above the top of the pulley casting *A*.

2. In tool-clamp *F*, place a roughing tool, and take a rough facing cut over the rim, feeding the tool toward the center.

3. In tool-clamp *G*, place a long, round-nosed, right-hand side tool. With hand feed, rough out one side and half the bottom of each channel. Replace this tool with a similar left-hand tool, and rough out the other side and half the bottom of each channel.

4. Reset the roughing tool in clamp *F*, placing it in a horizontal position. Take a down-feed cut over the outside of the casting, the cut being deep enough to clean up the crests on the grooved rim.

5. Replace the tool in clamp *G* with a rough forming tool, similar to *H*. Fasten this tool in a horizontal position, and feed it into one of the grooves until every part of the groove has been trued up. Before withdrawing the tool from the groove, make a depth mark on the face of the tool. This mark should coincide with the trued surfaces of the grooved rim. Machine each groove in the periphery of the rim, feeding the tool into each until the depth mark coincides with the trued surfaces. If a groove does not clean up when the tool has been fed to the depth indicated by the mark, it will be necessary to make the groove deeper, re-mark the tool, and machine all the other grooves to the same depth.

6. Replace the tool in clamp *F* with a bent roughing tool *K*, set horizontal, and face the bottom of the rim, working between the casting *A* and the table.

7. Replace the rough forming tool in clamp *G*, with a finishing tool *H*, which is ground to a gage. Repeat the operation described in step 5, and make all the channels conform to a gage.

8. Replace the tool in clamp *G* with a forming tool *L*. Set the tool in a horizontal position, and, feeding by hand, round off all the crests between the channels.

9. Place a broad-nosed finishing tool in clamp *F*, and a bent finishing tool, similar to *K*, in clamp *G*, and finish the top and bottom of the rim to dimension *a*. When taking these cuts a coarse feed should be used, and both the top and bottom of the rim faced simultaneously.

NOTE.—It will not be necessary to stop the machine at every change of tools.

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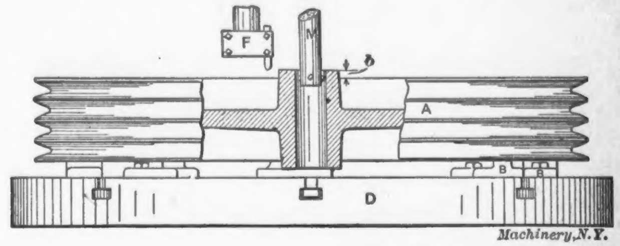
SHOP OPERATION SHEET NO. 51.

W. L. McLaren. MACHINERY, January, 1908.



To Bore and Face Hub of a Sheave Pulley Casting on a Vertical Boring Mill.

1. In the tool-clamp *F* place a round-nosed roughing tool, and face the top of the hub within 1/32 inch of the finish dimension *b*. If the hub is to be central with the rim, as is usually the case, the dimension *b* will equal one-half the difference between the distance through the hub and the width of the rim. When facing the hub, the dimension *b* is obtained by placing a straight-edge across the hub and measuring from the straight-edge to the rim.

2. Replace the roughing tool in clamp *F* with a flat finishing tool, and finish the top of the hub to dimension *b*, using a coarse feed.

3. Replace the tool-clamp in opposite head by a boring-bar *M*, provided with a single point cutter, set to fully clean the cored hole. Start the cut, feeding down. Test the hole with inside calipers for diameter at top and bottom. If these diameters are not equal, move the head slightly and take another light cut.

4. Replace the single point cutter with a double end cutter of proper size to bore the exact finish diameter, and take the cut, feeding down.

NOTE.—Boring bars, especially for large boring mills, are often made with an extension which passes down through a bushing in the table. This extension of the bar is a close sliding fit into the bushing, and prevents the bar from springing.

5. Release the casting *A* from the clamp blocks *B*, and turn it over; rest the rim on four to eight parallels, according to the diameter of the casting. Center it roughly by measuring from the edge of table *D*. Clamp to table with three to eight clamps and bolts, according to diameter of casting.

6. Face the hub as described in steps 1 and 2, and to conform to dimension, as at *b*.

NOTE.—If hole in hub is large enough, and the casting is set a sufficient height from the table *D* to permit it, an upwardly bent facing tool may be placed in the boring-bar, which is lowered through the hole for that purpose. The lower side of the hub can then be faced, provided there is room enough in the hole to allow the bar to move far enough in a horizontal direction. The length through the hub may be measured with a "hook scale." When the hub can be faced in this way, considerable time will be saved, as it will not be necessary to turn the casting over and reset it.

in this way, considerable time will be saved, as it will not be necessary to turn the casting over and reset it.

NOTE.—It will not be necessary to stop the machine at every change of tools.

a block or angle-piece bolted to the boring mill table, and against an arm, to act as a driver.

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I.—FORMULAS FOR HORSEPOWER TRANSMITTED BY CAST IRON AND RAWHIDE PINION

The tables of horsepowers transmitted by cast iron cut pinions are calculated by formulas derived from Reuleaux's "Constructor," using the following notation:

A = coefficient of wear
b = face or breadth of tooth
C = circular pitch
D = diameter of gear wheel, in inches
H = horsepower transmitted
N = revolutions per minute
P = pressure on tooth at pitch circle in pounds
R = pitch circle radius, in inches
S = permissible working stress of material of tooth, in pounds per square inch of section
V = velocity at pitch circle, in feet per minute,

then

$$H = \frac{bCRNS}{1058820} \qquad (1)$$

which is derived as follows:

A relation between the twisting moment PR and the corresponding horsepower is given by,

$$PR = 63025\frac{H}{N} \qquad (2)$$

The formula for strength is,

$$bC = 16.8\frac{P}{S} \qquad (3)$$

Combining (2) with (3) by substituting $\frac{63025H}{RN}$ for P in (3) gives $bC = \frac{1058820H}{RNS}$ which by transposition becomes (1)

The fiber stress S must be made less the higher the speed, and Reuleaux recommends for cast iron:

$$S = \frac{9600000}{V + 2164} \qquad (4)$$

which value was substituted in (1) for calculating the tables.

Allowance for wear is not provided in the tables, being left to the discretion of the designer. According to Reuleaux $A = \frac{PN}{b}$ should not exceed 28000

$$A = \frac{126050H}{Db}, \text{ or}$$

b should be greater than, or at least equal to, $4\frac{1}{2}\frac{H}{D}$.

In a pair of cast iron gears the greatest wear comes on the smallest gear. The coefficient A should not be greater than 28000 for the largest gear and may be taken as low as 12000 without obtaining inconvenient dimensions. To avoid excessive wear on the smaller gear, steel or bronze may be substituted.

The table of horsepowers transmitted by shrouded rawhide spur pinions of one inch face is calculated by the same formulas as were used for cast iron pinions, viz:

$$H = \frac{bCRNS}{1058820}$$

in which the notation is the same as before.

The permissible working stress per square inch of material is taken at 2520 pounds up to 2 diametral pitch, and for greater pitches it may be taken at the following values:

D.P.	$1\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{1}{4}$	1
S.	2340	2010	1670	1340

Owing to the elasticity of rawhide pinions, which enables them to sustain shocks without increase of stress, the working stress is made constant for the different pitches at all speeds. Hence the horsepower values in the tables are simple multiples of the first column, throughout.

Contributed by Joseph Holveck.

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II.—HORSEPOWER TRANSMITTED BY CAST IRON PINIONS.

		R. P. M.		100	150	200	250	300	350	400	450	500	550	600	650	700	750	800	850	950	1050	115
Pitch		Teeth	Pitch Diam.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.
5 D.P.	0.6283 C.P.	17	3.40	.424	.636	.827	1.01	1.19	1.38	1.54	1.70	1.85	2.00	2.14	2.28	2.45	2.58	2.70	2.83	3.05	3.26	3.5
		18	3.60	.453	.665	.866	1.07	1.26	1.44	1.61	1.80	1.96	2.11	2.27	2.41	2.55	2.68	2.82	2.94	3.18	3.40	3.6
		19	3.80	.478	.702	.915	1.12	1.32	1.51	1.69	1.86	2.03	2.19	2.35	2.50	2.64	2.79	2.92	3.05	3.30	3.53	3.7
		20	4.00	.503	.739	.964	1.18	1.39	1.59	1.78	1.96	2.14	2.31	2.43	2.58	2.74	2.88	3.02	3.16	3.42	3.66	3.8
		21	4.20	.528	.775	1.00	1.23	1.44	1.65	1.83	2.02	2.21	2.38	2.55	2.71	2.88	2.97	3.12	3.26	3.53	3.73	3.9
4 D.P.	0.7854 C.P.	17	4.25	.668	.969	1.26	1.62	1.82	2.07	2.31	2.55	2.79	3.01	3.22	3.43	3.57	3.76	3.95	4.13	4.46	4.72	5.0
		18	4.50	.701	1.03	1.34	1.70	1.91	2.19	2.45	2.66	2.85	3.13	3.35	3.58	3.78	3.92	4.11	4.30	4.66	5.00	5.1
		19	4.75	.740	1.08	1.40	1.72	2.00	2.26	2.54	2.81	3.06	3.24	3.49	3.90	3.92	4.07	4.27	4.46	4.77	5.12	5.4
		20	5.00	.779	1.13	1.48	1.78	2.09	2.36	2.67	2.90	3.16	3.36	3.60	3.83	4.06	4.21	4.42	4.63	5.02	5.26	5.5
		21	5.25	.808	1.18	1.53	1.86	2.19	2.45	2.76	3.04	3.26	3.54	3.78	3.96	4.13	4.42	4.58	4.72	5.12	5.52	5.7
$3\frac{1}{2}$ D.P.	0.8976 C.P.	17	4.86	.863	1.25	1.64	2.00	2.32	2.64	2.96	3.27	3.51	3.79	4.07	4.25	4.50	4.75	4.98	5.14	5.57	5.98	6.3
		18	5.14	.915	1.33	1.72	2.08	2.45	2.80	3.08	3.40	3.71	3.95	4.23	4.50	4.70	4.95	5.12	5.45	5.73	6.18	6.5
		19	5.43	.955	1.39	1.79	2.20	2.53	2.90	3.26	3.53	3.85	4.10	4.39	4.68	4.88	5.07	5.41	5.57	5.90	6.33	6.7
		20	5.72	1.00	1.46	1.89	2.31	2.67	3.05	3.36	3.65	3.99	4.31	4.55	4.77	5.05	5.34	5.52	5.70	6.21	6.48	6.9
		21	6.00	1.06	1.53	1.98	2.38	2.75	3.15	3.53	3.83	4.12	4.45	4.70	5.01	5.23	5.44	5.63	5.99	6.33	6.62	7.0
3 D.P.	1.0472 C.P.	17	5.66	1.16	1.69	2.18	2.67	3.09	3.47	3.88	4.29	4.62	4.99	5.26	5.60	5.94	6.17	6.38	6.59	7.18	7.82	7.9
		18	6.00	1.23	1.77	2.31	2.78	3.21	3.68	4.05	4.47	4.81	5.19	5.48	5.85	6.11	6.34	6.58	6.98	7.39	7.94	8.2
		19	6.33	1.28	1.87	2.41	2.94	3.38	3.80	4.27	4.65	4.98	5.39	5.69	5.98	6.25	6.69	6.94	7.19	7.59	8.15	8.5
		20	6.66	1.35	1.97	2.52	3.02	3.56	4.00	4.40	4.79	5.16	5.42	5.73	6.30	6.58	6.85	7.10	7.55	8.00	8.38	8.7
		21	7.00	1.42	2.05	2.64	3.17	3.68	4.13	4.57	5.05	5.42	5.77	6.11	6.41	6.71	7.01	7.26	7.71	8.16	8.58	9.1
$2\frac{1}{2}$ D.P.	1.2566 C.P.	17	6.80	1.65	2.38	3.08	3.70	4.29	4.90	5.40	5.88	6.31	6.72	7.11	7.47	8.05	8.38	8.71	8.98	9.50	10.00	10.
		18	7.20	1.73	2.52	3.23	3.85	4.44	5.10	5.64	6.11	6.58	7.00	7.53	7.91	8.30	8.65	8.95	9.25	9.81	10.30	11.
		19	7.60	1.83	2.63	3.38	4.06	4.69	5.28	5.84	6.35	6.83	7.29	7.71	8.11	8.52	8.86	9.19	9.50	10.10	10.88	
		20	8.00	1.93	2.74	3.56	4.27	4.85	5.48	6.03	6.57	7.07	7.67	7.89	8.32	8.70	9.07	9.42	9.76	10.37	10.90	
		21	8.40	2.00	2.88	3.67	4.41	5.09	5.75	6.33	6.79	7.32	7.81	8.28	8.75	9.14	9.26	9.64	9.99	10.88		

Contributed by Joseph Holveck.

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.—FORMULAS FOR HORSEPOWER TRANSMITTED BY CAST IRON AND RAWHIDE PINIONS.

The tables of horsepowers transmitted by cast iron cut pinions are calculated by formulas derived from Reuleaux's "Constructor," using the following notation:

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- C = circular pitch
- D = diameter of gear wheel, in inches
- H = horsepower transmitted
- N = revolutions per minute
- P = pressure on tooth at pitch circle in pounds
- R = pitch circle radius, in inches
- S = permissible working stress of material of tooth, in pounds per square inch of section
- V = velocity at pitch circle, in feet per minute,

then

$$H = \frac{bCRNS}{1058820} \qquad (1)$$

which is derived as follows:

A relation between the twisting moment PR and the corresponding horsepower is given by,

$$PR = 63025\frac{H}{N} \qquad (2)$$

The formula for strength is,

$$bC = 16.8\frac{P}{S} \qquad (3)$$

Combining (2) with (3) by substituting $\frac{63025H}{RN}$ for P in (3) gives $bC = \frac{1058820H}{RNS}$ which by transposition becomes (1)

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$$A = \frac{126050H}{Db}, \text{ or}$$

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In a pair of cast iron gears the greatest wear comes on the smallest gear. The coefficient A should not be greater than 28000 for the largest gear and may be taken as low as 12000 without obtaining inconvenient dimensions. To avoid excessive wear on the smaller gear, steel or bronze may be substituted.

The table of horsepowers transmitted by shrouded rawhide spur pinions of one inch face is calculated by the same formulas as were used for cast iron pinions, viz:

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in which the notation is the same as before.

The permissible working stress per square inch of material is taken at 2520 pounds up to 2 diametral pitch, and for greater pitches it may be taken at the following values:

D.P.	$1\frac{3}{4}$	$1\frac{1}{2}$	$1\frac{1}{4}$	1
S.	2340	2010	1670	1340

Owing to the elasticity of rawhide pinions, which enables them to sustain shocks without increase of stress, the working stress is made constant for the different pitches at all speeds. Hence the horsepower values in the tables are simple multiples of the first column, throughout.

ontributed by Joseph Holveck.

II.—HORSEPOWER TRANSMITTED BY CAST IRON PINIONS.

		R. P. M.		100	150	200	250	300	350	400	450	500	550	600	650	700	750	800	850	950	1050	1150
Pitch		Teeth	Pitch Diam.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.	H.P.
5 D.P.	0.6283 C.P.	17	3.40	.424	.636	.827	1.01	1.19	1.38	1.54	1.70	1.85	2.00	2.14	2.28	2.45	2.58	2.70	2.83	3.05	3.26	3.51
		18	3.60	.453	.665	.866	1.07	1.26	1.44	1.61	1.80	1.96	2.11	2.27	2.41	2.55	2.68	2.82	2.94	3.18	3.40	3.60
		19	3.80	.478	.702	.915	1.12	1.32	1.51	1.69	1.86	2.03	2.19	2.35	2.50	2.64	2.79	2.92	3.05	3.30	3.53	3.70
		20	4.00	.503	.739	.964	1.18	1.39	1.59	1.78	1.96	2.14	2.31	2.43	2.58	2.74	2.88	3.02	3.16	3.42	3.66	3.89
		21	4.20	.528	.775	1.00	1.23	1.44	1.65	1.83	2.02	2.21	2.38	2.55	2.71	2.88	2.97	3.12	3.26	3.53	3.73	3.97
4 D.P.	0.7854 C.P.	17	4.25	.668	.969	1.26	1.62	1.82	2.07	2.31	2.55	2.79	3.01	3.22	3.43	3.57	3.76	3.95	4.13	4.46	4.72	5.02
		18	4.50	.701	1.03	1.34	1.70	1.91	2.19	2.45	2.66	2.85	3.13	3.35	3.58	3.78	3.92	4.11	4.30	4.66	5.00	5.18
		19	4.75	.740	1.08	1.40	1.72	2.00	2.26	2.54	2.81	3.06	3.24	3.49	3.90	3.92	4.07	4.27	4.46	4.77	5.12	5.47
		20	5.00	.779	1.13	1.48	1.78	2.09	2.36	2.67	2.90	3.16	3.36	3.60	3.83	4.06	4.21	4.42	4.63	5.02	5.26	5.59
		21	5.25	.808	1.18	1.53	1.86	2.19	2.45	2.76	3.04	3.26	3.54	3.78	3.96	4.13	4.42	4.58	4.72	5.12	5.52	5.71
$3\frac{1}{2}$ D.P.	0.8976 C.P.	17	4.86	.863	1.25	1.64	2.00	2.32	2.64	2.96	3.27	3.51	3.79	4.07	4.25	4.50	4.75	4.98	5.14	5.57	5.98	6.38
		18	5.14	.915	1.33	1.72	2.08	2.45	2.80	3.08	3.40	3.71	3.95	4.23	4.50	4.70	4.95	5.12	5.45	5.73	6.18	6.56
		19	5.43	.955	1.39	1.79	2.20	2.53	2.90	3.26	3.53	3.85	4.10	4.39	4.68	4.88	5.07	5.41	5.57	5.90	6.33	6.75
		20	5.72	1.00	1.46	1.89	2.31	2.67	3.05	3.36	3.65	3.99	4.31	4.55	4.77	5.05	5.34	5.52	5.70	6.21	6.48	6.90
		21	6.00	1.06	1.53	1.98	2.38	2.75	3.15	3.53	3.83	4.12	4.45	4.70	5.01	5.23	5.44	5.63	5.99	6.33	6.62	7.08
3 D.P.	1.0472 C.P.	17	5.66	1.16	1.69	2.18	2.67	3.09	3.47	3.88	4.29	4.62	4.99	5.26	5.60	5.94	6.17	6.38	6.59	7.18	7.82	7.98
		18	6.00	1.23	1.77	2.31	2.78	3.21	3.68	4.05	4.47	4.81	5.19	5.48	5.85	6.11	6.34	6.58	6.98	7.39	7.94	8.26
		19	6.33	1.28	1.87	2.41	2.94	3.38	3.80	4.27	4.65	4.98	5.39	5.69	5.98	6.25	6.69	6.94	7.19	7.59	8.15	8.50
		20	6.66	1.35	1.97	2.52	3.02	3.56	4.00	4.40	4.79	5.16	5.42	5.73	6.30	6.58	6.85	7.10	7.55	8.00	8.38	8.72
		21	7.00	1.42	2.05	2.64	3.17	3.68	4.13	4.57	5.05	5.42	5.77	6.11	6.41	6.71	7.01	7.26	7.71	8.16	8.58	9.16
$2\frac{1}{2}$ D.P.	1.2566 C.P.	17	6.80	1.65	2.38	3.08	3.70	4.29	4.90	5.40	5.88	6.31	6.72	7.11	7.47	8.05	8.38	8.71	8.98	9.50	10.00	10.67
		18	7.20	1.73	2.52	3.23	3.85	4.44	5.10	5.64	6.11	6.58	7.00	7.53	7.91	8.30	8.65	8.95	9.25	9.81	10.30	11.05
		19	7.60	1.83	2.63	3.38	4.06	4.69	5.28	5.84	6.35	6.83	7.29	7.71	8.11	8.52	8.86	9.19	9.50	10.10	10.88	
		20	8.00	1.93	2.74	3.56	4.27	4.85	5.48	6.03	6.57	7.07	7.67	7.89	8.32	8.70	9.07	9.42	9.76	10.37	10.90	
		21	8.40	2.00	2.88	3.67	4.41	5.09	5.75	6.33	6.79	7.32	7.81	8.28	8.75	9.14	9.26	9.64	9.99	10.88		

ontributed by Joseph Holveck.

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This data sheet is made up so as to be readily bound. It may be cut into four sections, 6x9 inches in size, and bound into note-book form for convenient reference, by means of staples inserted into holes punched at points indicated.

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III.—HORSEPOWER TRANSMITTED BY CAST IRON PINIONS.

Pitch		Teeth	Pitch Diam.	100 H.P.	150 H.P.	200 H.P.	250 H.P.	300 H.P.	350 H.P.	400 H.P.	450 H.P.	500 H.P.	550 H.P.	600 H.P.	650 H.P.	700 H.P.	750 H.P.	800 H.P.	850 H.P.	950 H.P.	1050 H.P.	1150 H.P.
2 D.P.	1.5708 C.P.	17	8.50	2.53	3.61	4.64	5.58	6.44	7.15	7.89	8.59	9.26	9.88	10.47	10.73	10.94	11.43	11.90	12.64	13.72		
		18	9.00	2.66	3.82	4.80	5.79	6.71	7.57	8.22	8.95	9.51	10.16	10.81	11.36	11.92	12.10	12.60	13.05			
		19	9.50	2.80	4.00	5.07	6.11	6.97	7.84	8.54	9.32	10.04	10.40	11.07	11.67	12.23	12.47	12.90				
		20	10.00	2.95	4.17	5.34	6.32	7.21	8.12	8.84	9.51	10.27	11.01	11.36	11.95	12.56	12.80	13.64				
		21	10.50	3.07	4.38	5.51	6.52	7.42	8.39	9.15	9.98	10.51	11.21	11.58	12.24	12.86	13.43					
$1\frac{3}{4}$ D.P.	1.7952 C.P.	17	9.71	3.28	4.67	5.93	7.02	8.15	9.02	9.98	10.90	11.40	12.23	12.95	13.28	13.95	14.57	15.15				
		18	10.30	3.43	4.90	6.17	7.43	8.47	9.39	10.25	11.17	12.07	12.55	13.33	14.05	14.40	15.03					
		19	10.86	3.48	5.06	6.51	7.73	8.77	9.75	10.82	11.46	12.41	12.90	13.69	14.46	14.80						
		20	11.43	3.78	5.33	6.72	7.99	9.10	10.10	11.05	12.08	12.70	13.59	14.07	14.86							
		21	12.00	3.91	5.60	6.94	8.25	9.40	10.42	11.27	12.36	12.97	14.27	14.40	15.20							
$1\frac{1}{2}$ D.P.	2.0944 C.P.	17	11.33	4.37	6.17	7.39	9.24	10.52	11.69	12.78	13.62	14.68	15.72	16.27	17.20	18.05						
		18	12.00	4.57	6.53	8.08	9.61	10.96	12.21	13.15	14.41	15.14	16.18	16.80	17.76							
		19	12.67	4.82	6.76	8.54	10.14	11.42	12.50	13.87	14.76	15.65	16.66	17.28								
		20	13.33	5.04	7.13	8.84	10.49	11.80	13.16	14.25	15.14	16.36	17.13	18.21								
		21	14.00	5.23	7.35	9.14	10.83	12.21	13.42	14.50	15.45	16.75	17.50									
$1\frac{1}{4}$ D.P.	2.5133 C.P.	17	13.60	6.17	8.57	10.81	12.63	14.23	16.10	17.43	18.51	19.53	20.42									
		18	14.40	6.46	9.23	11.28	13.16	15.07	16.57	17.91	19.07	20.17	21.62									
		19	15.20	6.76	9.36	11.69	13.66	15.42	17.05	18.40	19.65	20.75										
		20	16.00	7.12	9.74	12.30	14.38	15.78	17.41	18.83	20.08	21.86										
		21	16.80	7.32	10.20	12.48	14.66	16.57	18.29	19.30	20.64											
1 D.P.	3.1416 C.P.	17	17.00	9.26	12.90	15.79	18.68	20.96	22.50	24.40	26.10											
		18	18.00	9.80	13.42	16.45	19.02	21.63	23.82	25.20	27.86											
		19	19.00	10.34	13.95	17.08	20.08	22.15	24.46	25.90												
		20	20.00	10.68	14.42	17.68	20.55	22.70	25.13	27.30												
		21	21.00	11.03	14.86	18.32	21.03	23.83	25.73													

Contributed by Joseph Holveck.

CREASE HERE

IV.—HORSEPOWER TRANSMITTED BY RAWHIDE PINIONS.

Pitch		Teeth	Pitch Diam.	100 H.P.	150 H.P.	200 H.P.	250 H.P.	300 H.P.	350 H.P.	400 H.P.	450 H.P.
5 D.P.	0.6283 C.P.	17	3.40	.254	.381	.508	.635	.763	.889	1.02	1.14
		18	3.60	.269	.404	.538	.673	.807	.942	1.07	1.21
		19	3.80	.284	.426	.568	.710	.852	.994	1.13	1.28
		20	4.00	.299	.449	.598	.748	.897	1.04	1.19	1.34
		21	4.20	.314	.471	.628	.785	.942	1.10	1.25	1.41
4 D.P.	0.7854 C.P.	17	4.25	.397	.596	.794	.992	1.19	1.39	1.59	1.79
		18	4.50	.420	.631	.841	1.05	1.26	1.47	1.68	1.89
		19	4.75	.444	.666	.888	1.04	1.33	1.55	1.77	1.99
		20	5.00	.467	.701	.934	1.17	1.40	1.63	1.87	2.10
		21	5.25	.491	.736	.981	1.23	1.47	1.72	1.96	2.21
$3\frac{1}{2}$ D.P.	0.8976 C.P.	17	4.86	.518	.777	1.04	1.29	1.55	1.81	2.07	2.33
		18	5.14	.549	.824	1.10	1.37	1.65	1.92	2.19	2.47
		19	5.43	.580	.870	1.16	1.45	1.74	2.03	2.32	2.61
		20	5.72	.610	.914	1.22	1.52	1.83	2.13	2.44	2.74
		21	6.00	.641	.961	1.28	1.60	1.92	2.24	2.56	2.88
3 D.P.	1.0472 C.P.	17	5.66	.705	1.06	1.41	1.76	2.12	2.47	2.82	3.17
		18	6.00	.748	1.12	1.50	1.87	2.24	2.62	2.99	3.36
		19	6.33	.789	1.18	1.58	1.97	2.37	2.76	3.16	3.55
		20	6.66	.831	1.25	1.66	2.08	2.49	2.91	3.32	3.74
		21	7.00	.872	1.31	1.74	2.18	2.62	3.05	3.49	3.93
$2\frac{1}{2}$ D.P.	1.2566 C.P.	17	6.80	1.02	1.52	2.03	2.54	3.05	3.56	4.06	4.57
		18	7.20	1.08	1.61	2.15	2.69	3.23	3.76	4.30	4.84
		19	7.60	1.14	1.70	2.27	2.84	3.41	3.97	4.54	5.11
		20	8.00	1.20	1.79	2.39	2.99	3.59	4.18	4.78	5.38
		21	8.40	1.26	1.88	2.51	3.14	3.77	4.39	5.02	5.65

Pitch		Teeth	Pitch Diam.	100 H.P.	150 H.P.	200 H.P.	250 H.P.	300 H.P.	350 H.P.	400 H.P.	450 H.P.
2 D.P.	1.5708 C.P.	17	8.50	1.59	2.38	3.17	3.97	4.76	5.56	6.35	7.15
		18	9.00	1.68	2.52	3.36	4.20	5.04	5.88	6.72	7.57
		19	9.50	1.78	2.66	3.55	4.44	5.32	6.21	7.10	7.99
		20	10.00	1.87	2.80	3.74	4.67	5.60	6.54	7.47	8.40
		21	10.50	1.96	2.94	3.92	4.90	5.89	6.87	7.85	8.83
$1\frac{3}{4}$ D.P.	1.7952 C.P.	17	9.71	1.93	2.89	3.85	4.82	5.78	6.74	7.71	8.67
		18	10.3	2.04	3.06	4.08	5.10	6.12	7.14	8.16	9.18
		19	10.86	2.15	3.23	4.30	5.38	6.46	7.53	8.61	9.68
		20	11.43	2.27	3.40	4.54	5.67	6.80	7.94	9.07	10.20
		21	12.00	2.38	3.57	4.76	5.95	7.14	8.33	9.52	10.71
$1\frac{1}{2}$ D.P.	2.0944 C.P.	17	11.33	2.25	3.38	4.51	5.63	6.76	7.89	9.01	10.14
		18	12.00	2.39	3.58	4.77	5.96	7.16	8.35	9.54	10.73
		19	12.67	2.52	3.77	5.03	6.29	7.55	8.81	10.07	11.32
		20	13.33	2.65	3.98	5.30	6.63	7.96	9.28	10.61	11.94
		21	14.00	2.78	4.17	5.56	6.96	8.35	9.74	11.13	12.52
$1\frac{1}{4}$ D.P.	2.5133 C.P.	17	13.60	2.70	4.04	5.39	6.74	8.09	9.43	10.78	12.13
		18	14.40	2.85	4.28	5.71	7.13	8.56	9.88	11.42	12.84
		19	15.20	3.01	4.52	6.02	7.53	9.04	10.54	12.05	13.55
		20	16.00	3.17	4.76	6.34	7.93	9.51	11.10	12.68	14.27
		21	16.80	3.33	4.99	6.66	8.32	9.99	11.65	13.32	14.98
1 D.P.	3.1416 C.P.	17	17.00	3.38	5.07	6.76	8.45	10.14	11.83	13.52	15.21
		18	18.00	3.58	5.37	7.16	8.95	10.73	12.52	14.31	16.10
		19	19.00	3.78	5.66	7.55	9.44	11.33	13.22	15.10	
		20	20.00	3.98	5.96	7.95	9.94	11.93	13.91	15.90	
		21	21.00	4.17	6.26	8.35	10.44	12.53	14.61		

Contributed by Joseph Holveck.